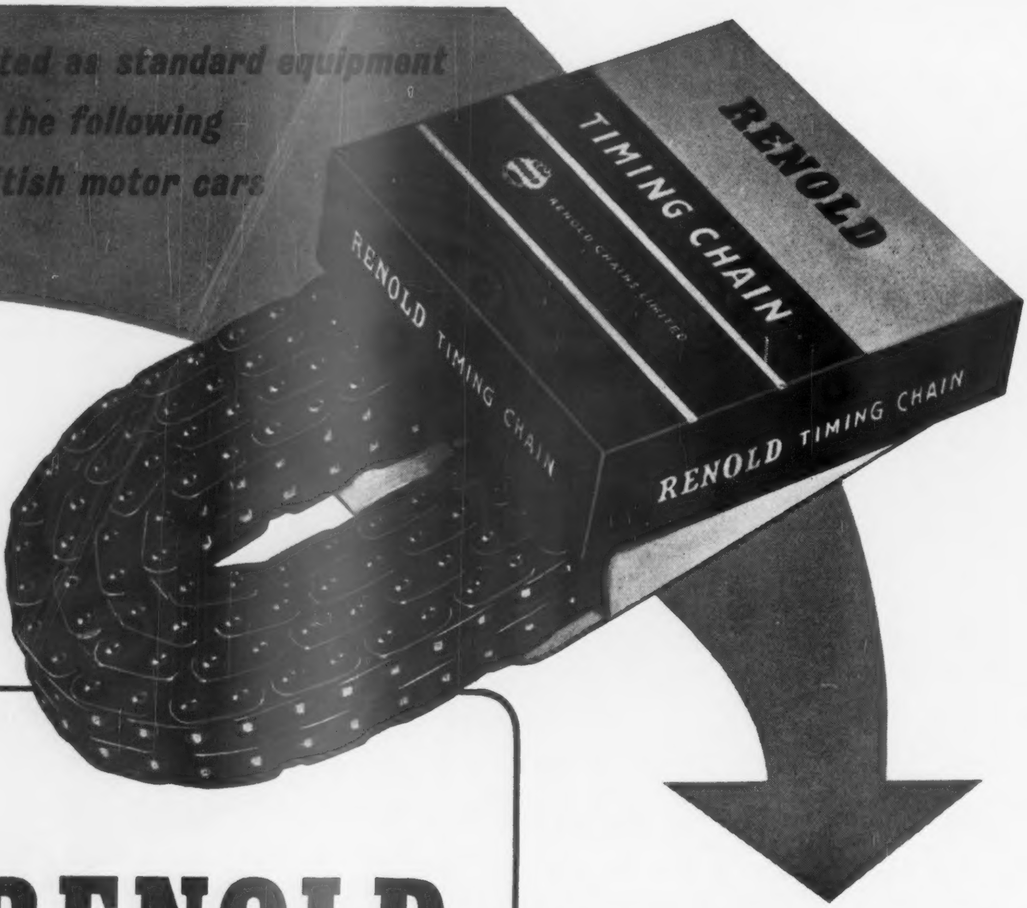
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 47 No. 8 AUGUST 1957 PRICE: 3s. 6d.

Fitted as standard equipment on the following British motor cars



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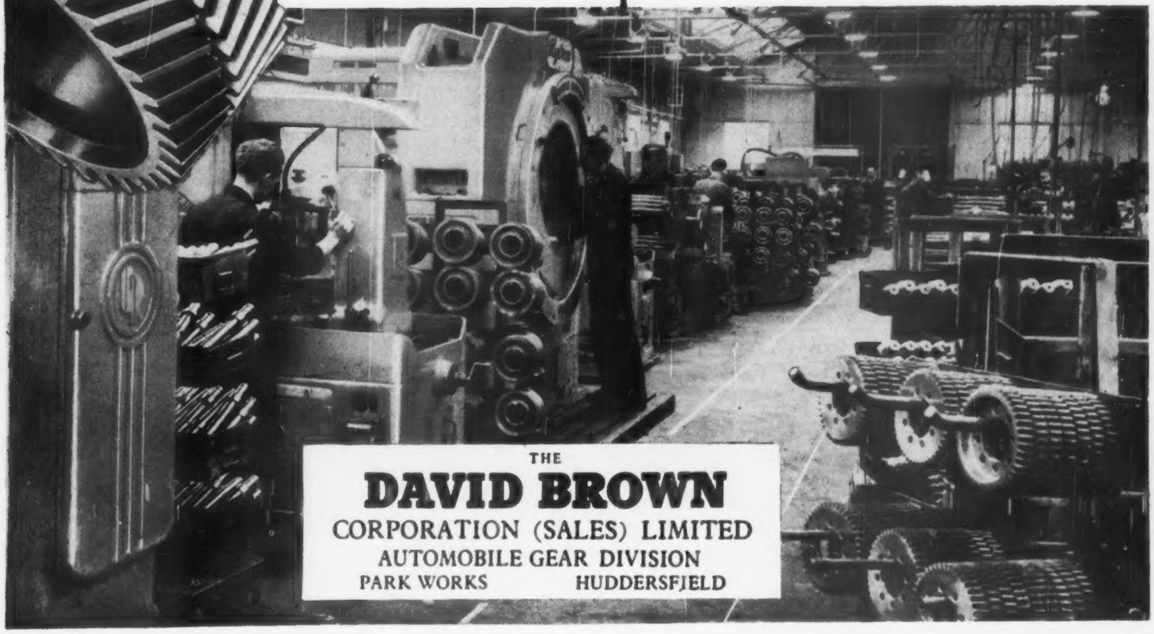

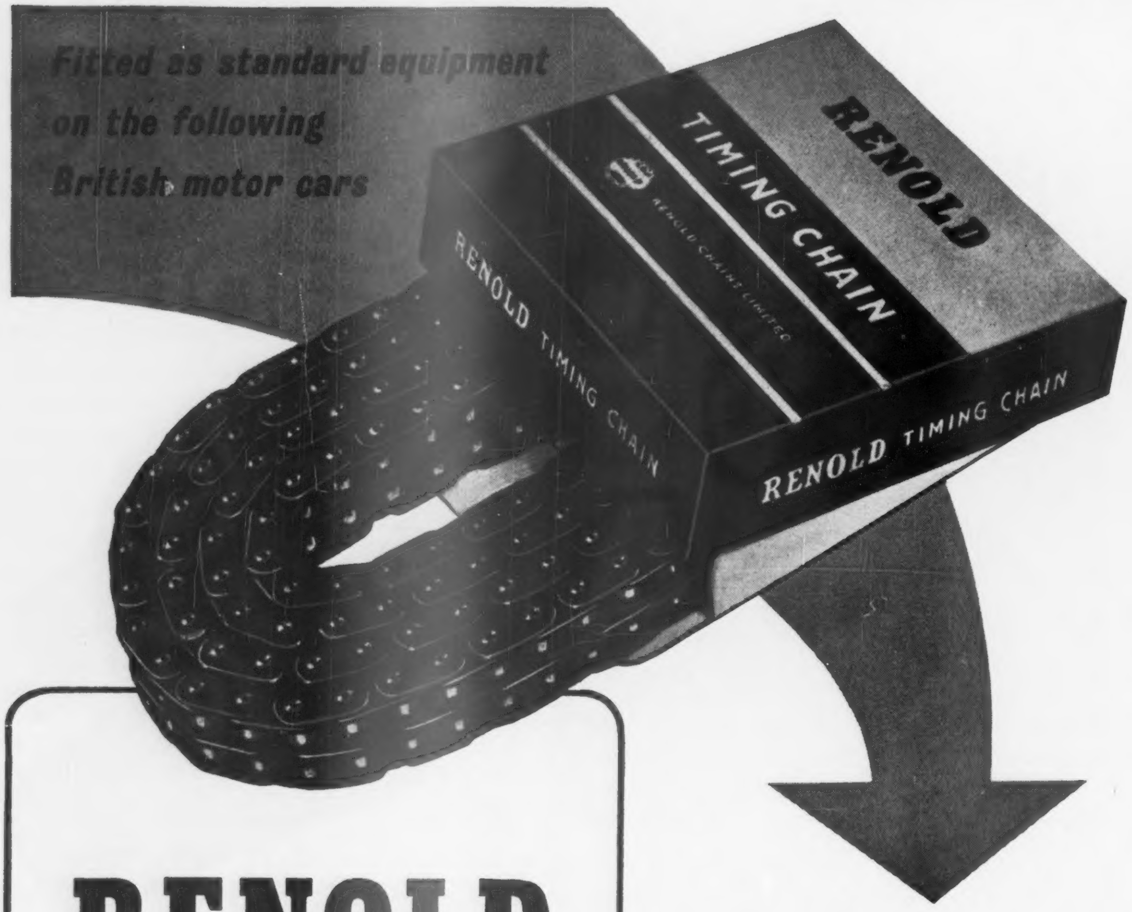

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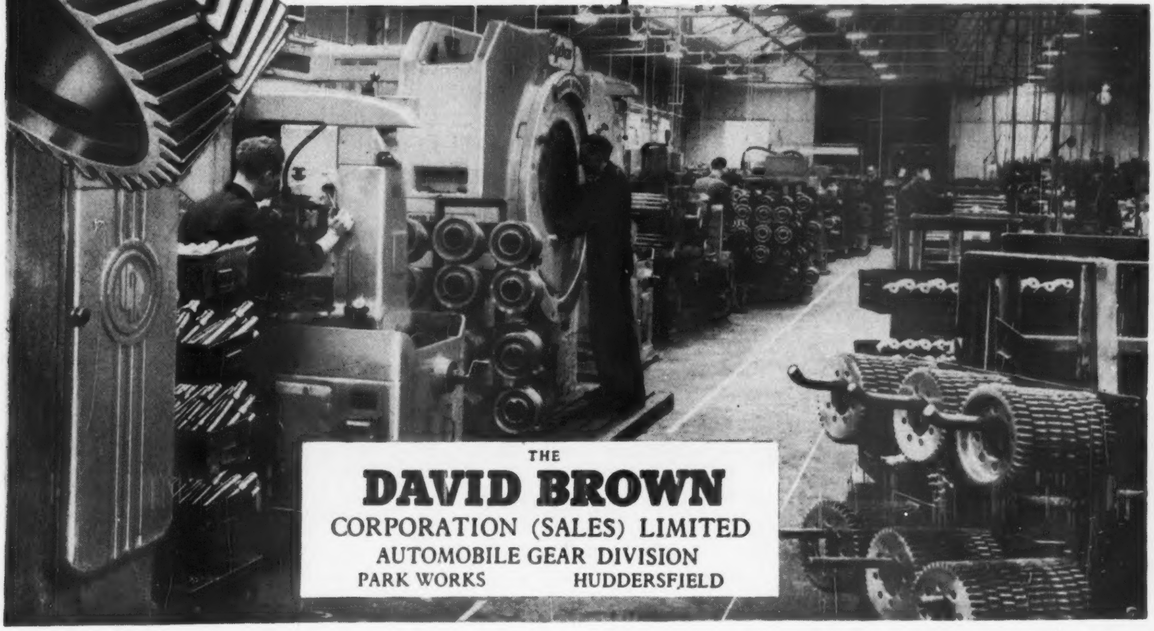

THE
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CORPORATION (SALES) LIMITED
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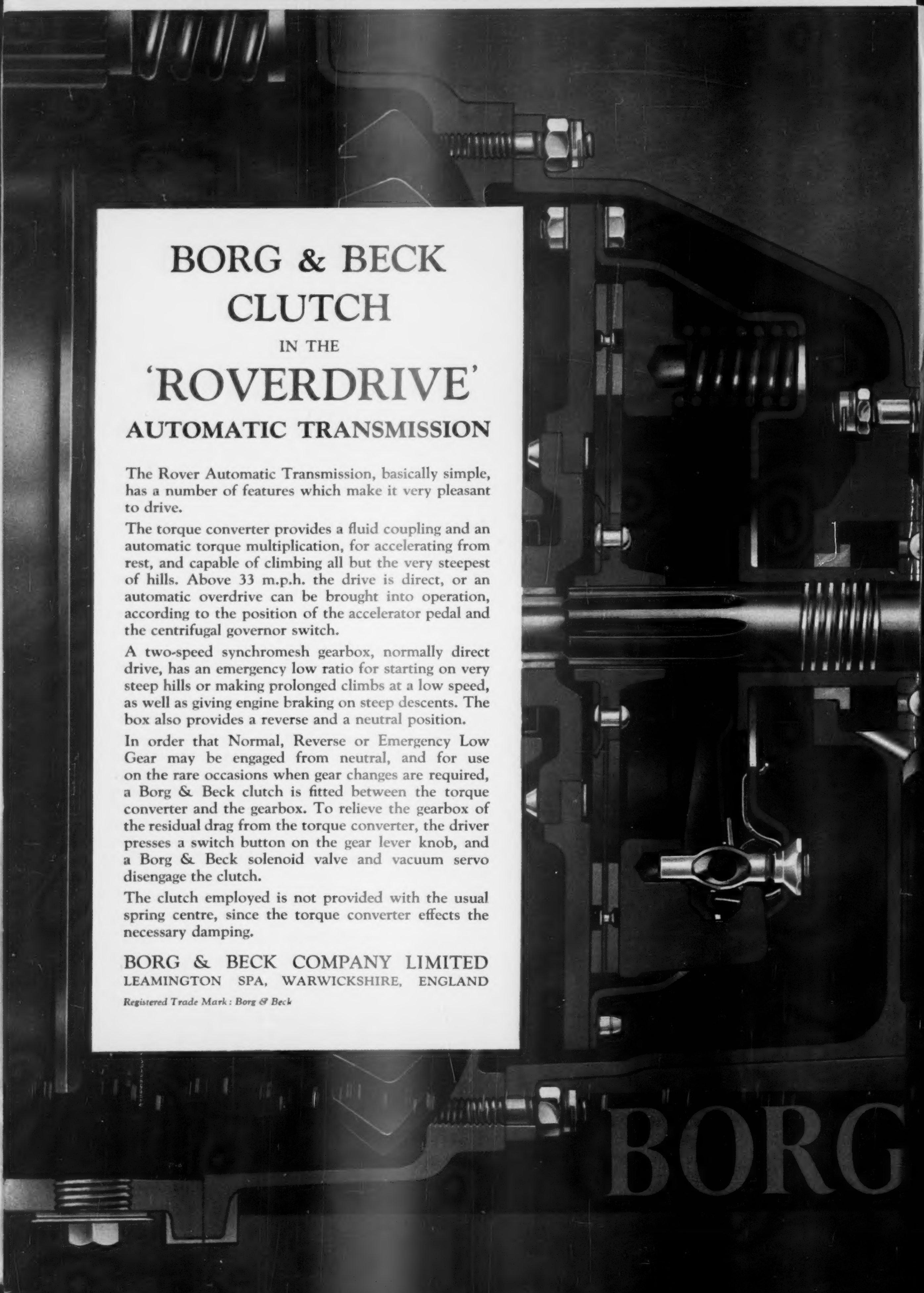
PARK WORKS HUDDERSFIELD


BORG & BECK
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IN THE
'ROVERDRIVE'
AUTOMATIC TRANSMISSION
The Rover Automatic Transmission, basically simple, has a number of features which make it very pleasant to drive.
The torque converter provides a fluid coupling and an automatic torque multiplication, for accelerating from rest, and capable of climbing all but the very steepest of hills. Above 33 m.p.h. the drive is direct, or an automatic overdrive can be brought into operation, according to the position of the accelerator pedal and the centrifugal governor switch.
A two-speed synchromesh gearbox, normally direct drive, has an emergency low ratio for starting on very steep hills or making prolonged climbs at a low speed, as well as giving engine braking on steep descents. The box also provides a reverse and a neutral position.
In order that Normal, Reverse or Emergency Low Gear may be engaged from neutral, and for use on the rare occasions when gear changes are required, a Borg & Beck clutch is fitted between the torque converter and the gearbox. To relieve the gearbox of the residual drag from the torque converter, the driver presses a switch button on the gear lever knob, and a Borg & Beck solenoid valve and vacuum servo disengage the clutch.
The clutch employed is not provided with the usual spring centre, since the torque converter effects the necessary damping.
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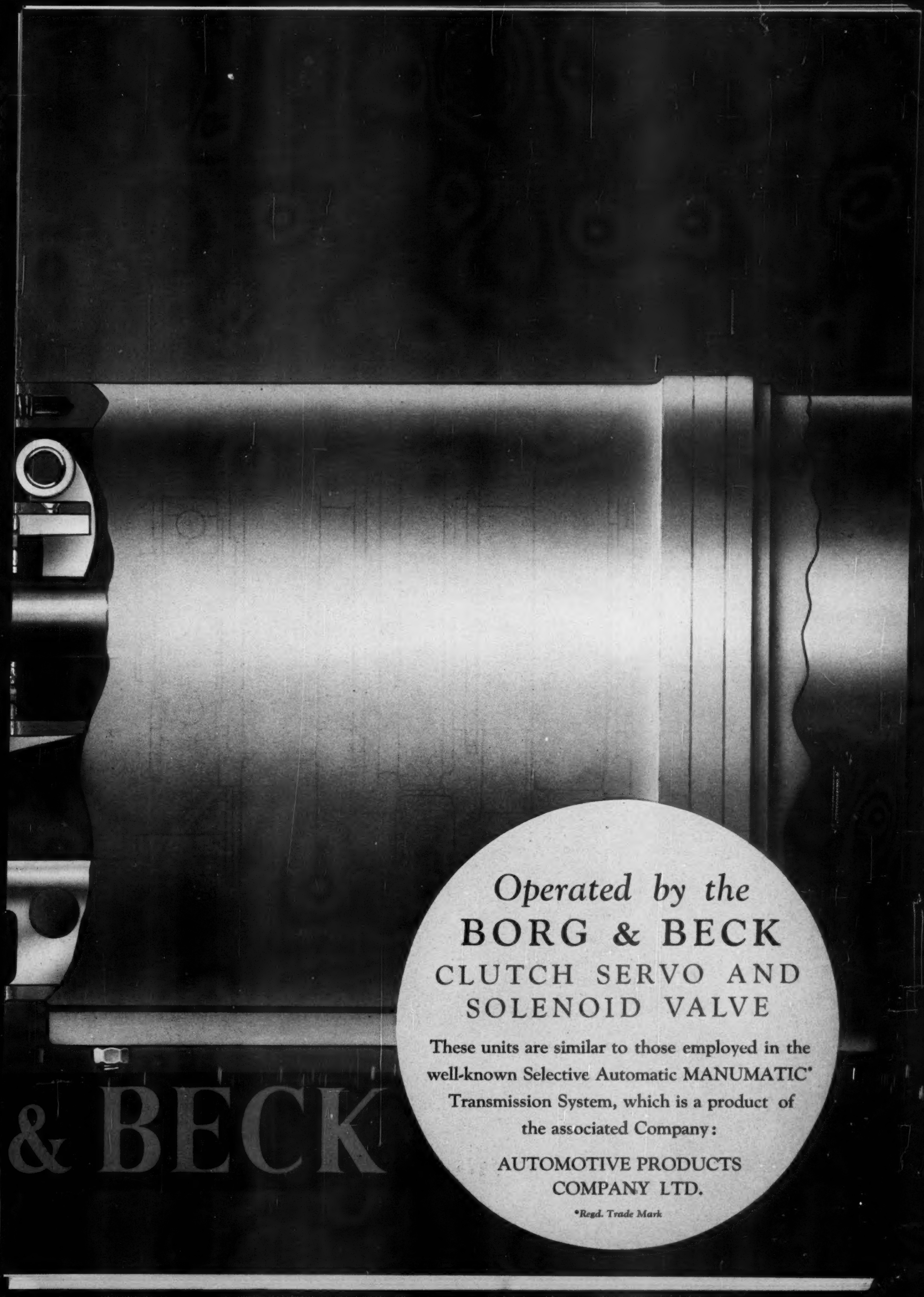
Registered Trade Mark: Borg & Beck
BORG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Operated by the
BORG & BECK
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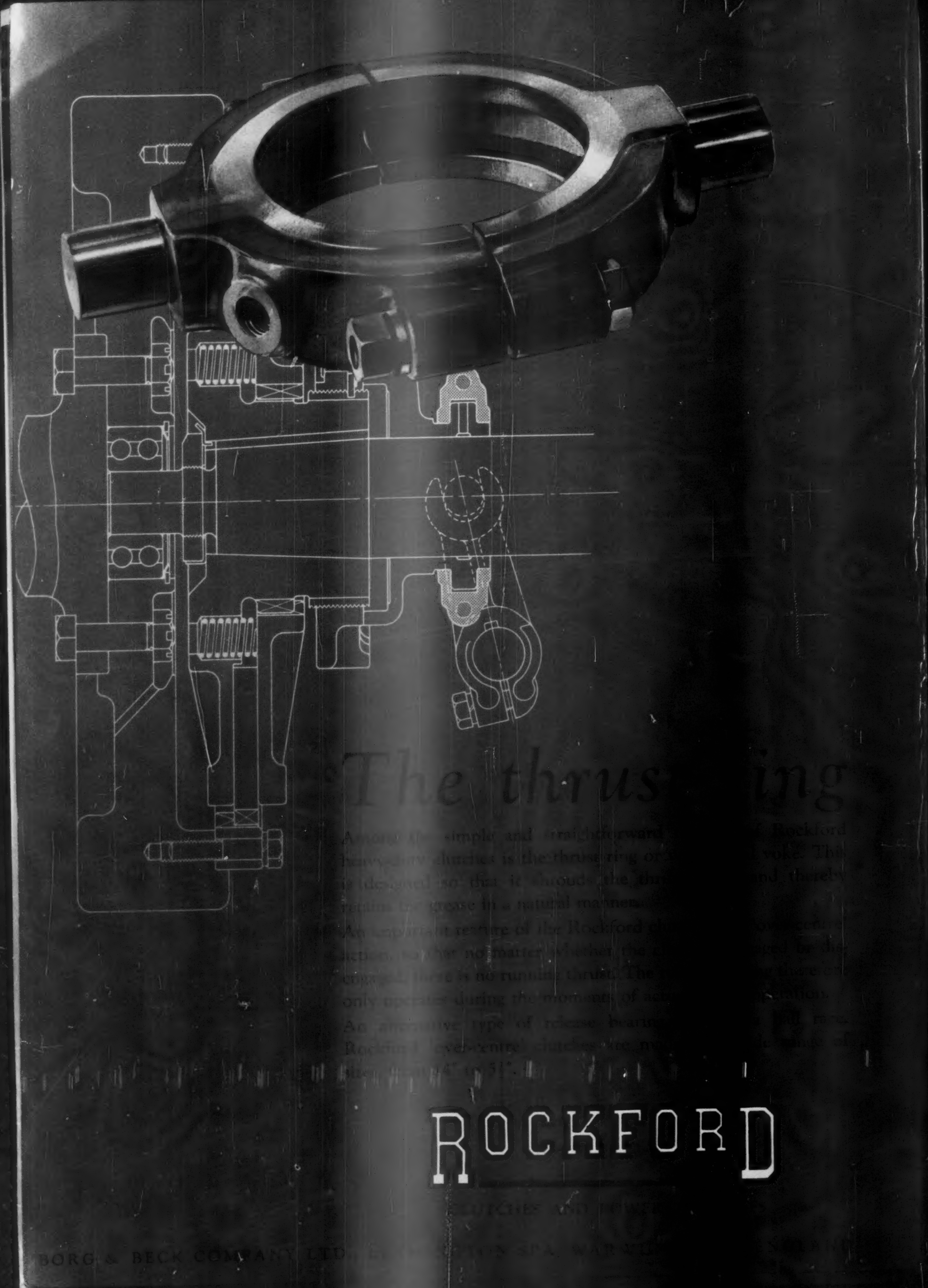
SOLENOID VALVE
These units are similar to those employed in the well-known Selective Automatic MANUMATIC* Transmission System, which is a product of the associated Company:
AUTOMOTIVE PRODUCTS
COMPANY LTD.
*Regd. Trade Mark



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Among the simple and straightforward
heavy-duty clutches is the thrust ring or
is designed so that it shrouds the
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DESOUTTER improves even on perfect tightness..!

It's PNEU! It's the PNEU pneumatic lightning tightener... Desoutter's amazing PNEU Multiple Nut Runner— the P.M.N.R. It screws up not one nut... not two nuts... not three nuts... but *several* nuts. All together. Fast. Just like that. And what's more—this latest model gives nuts a startling new tightness they've never felt before!

WHAT! even on perfect tightness..?

Here's how. First—a fast run at a low torque. Then—an *automatic* change to slow speed to tighten the nuts to torques of up to 135 lb. ft. It's true friends... once Desoutter's P.M.N.R. gets going, all work vanishes. No effort. No brain strain... and it's *incredibly* kind to the hands.

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Prove it for yourself today.
Lug that job off the assembly line
and over to the window.
There in that critical, searching light
you'll see the kind of hair-splitting
tightness that tires the eyes.
Desoutter turns *tightest of all!*

DESOUTTER

We thought we'd better ask the M.D. what he thought of our new-style advertising. "I don't quite understand," he smugged. "It sounds as if we're trying to *sell* Desoutter tools." As if he never did!

With acknowledgments to the makers of Omo, Daz, Surf, Persil, Tide, Dreft, Fab, Cheer and all others of that inimitable school.

squashes tighter

DESOUTTER BROS. LIMITED, THE HYDE, HENDON, N.W.9. TEL: COLINDALE 6346 (5 LINES) GRAMS: DESPNUCO, HYDE, LONDON

CRC 284

OPEN THE DOOR TO 'MG 5'

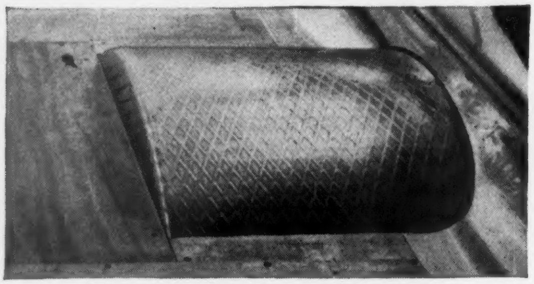
REGD. TRADE MARK

At the entrance to the 41-seater Duple 'Elizabethan', the bright appearance of the 'MG5' chequer plate steps gives the passenger a good impression. 'MG5' is also favoured by coach operators because this tough hard-wearing alloy takes all the rough treatment that comes its way—kicks, scraping, the grinding of mud and gravel underfoot, and exposure to water. Then, with a quick wash down, it comes up shining and clean again. 'MG5' never rusts, never needs painting and is only one-third the weight of steel. Better appearance, harder wear, less maintenance, lighter weight—are all good reasons for choosing 'MG5' for stairs, decking, wheel arch covers and other hard-wearing coach parts.

Our alloys, which include the famous 'Duralumin' series, are used for many purposes in road transport. If you would like advice on how aluminium alloys can help you, please let us know. Our Technical Sales Section is always ready to help.

The illustration at left shows 'MG5' chequer plate used at a coach entrance.
Below. Wheel arches that will never rust! 'MG5' is ideal for jobs like this!

James Booth



JAMES BOOTH & COMPANY LIMITED · ARGYLE ST. WORKS · BIRMINGHAM · 7

Extrusions, large forgings, plate, sheet, strip, tubes and wire in brass and copper, as well as light alloys

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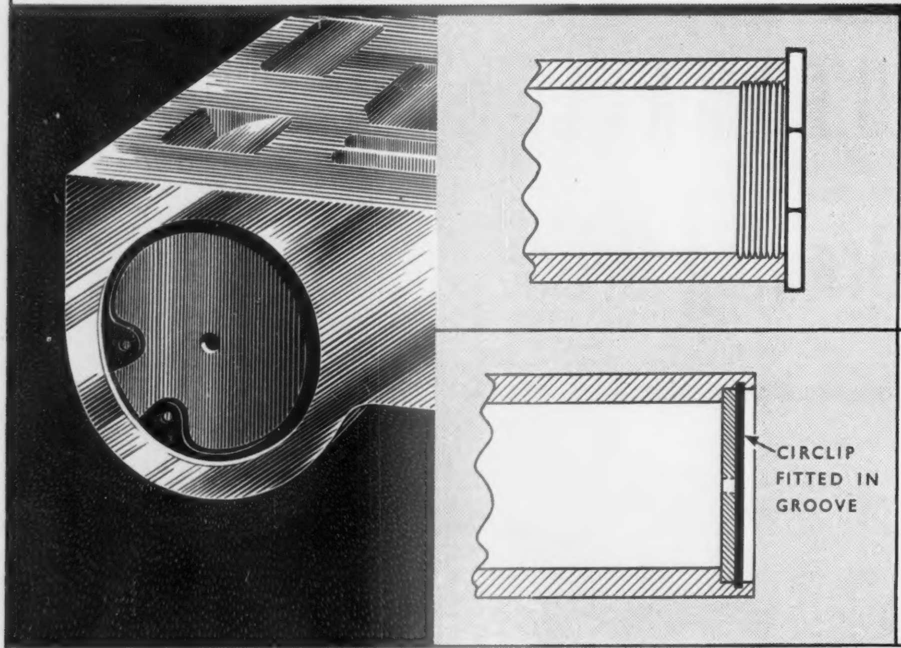
The logical advance in Retaining



OLD WAY

This fluid seal involved internal threading of the tube, which was sealed with an expensive cap-nut. The assembly was laborious and spanners were needed.

THE SALTER WAY

The tube is recessed and then simply grooved with the SALTER Grooving Tool. A Circlip is snapped into position and secures the fluid retaining plate with positive, vibration-free locking. When necessary the Circlip can be removed quickly and easily.

save material–reduce assembly time – cut costs

When it's a question of assembling components in any engineering field, Salter Retainers are the answer. They replace nuts and bolts, screws, cotter pins, and eliminate expensive threading and machining operations. A large standard range is at your immediate disposal, and we should welcome the opportunity to assist in developing special retainers to solve your problems.

Send for the Salter Retainer catalogue — no designer is complete without it.

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Geo. Salter & Co. Ltd., West Bromwich · Spring Specialists since 1760

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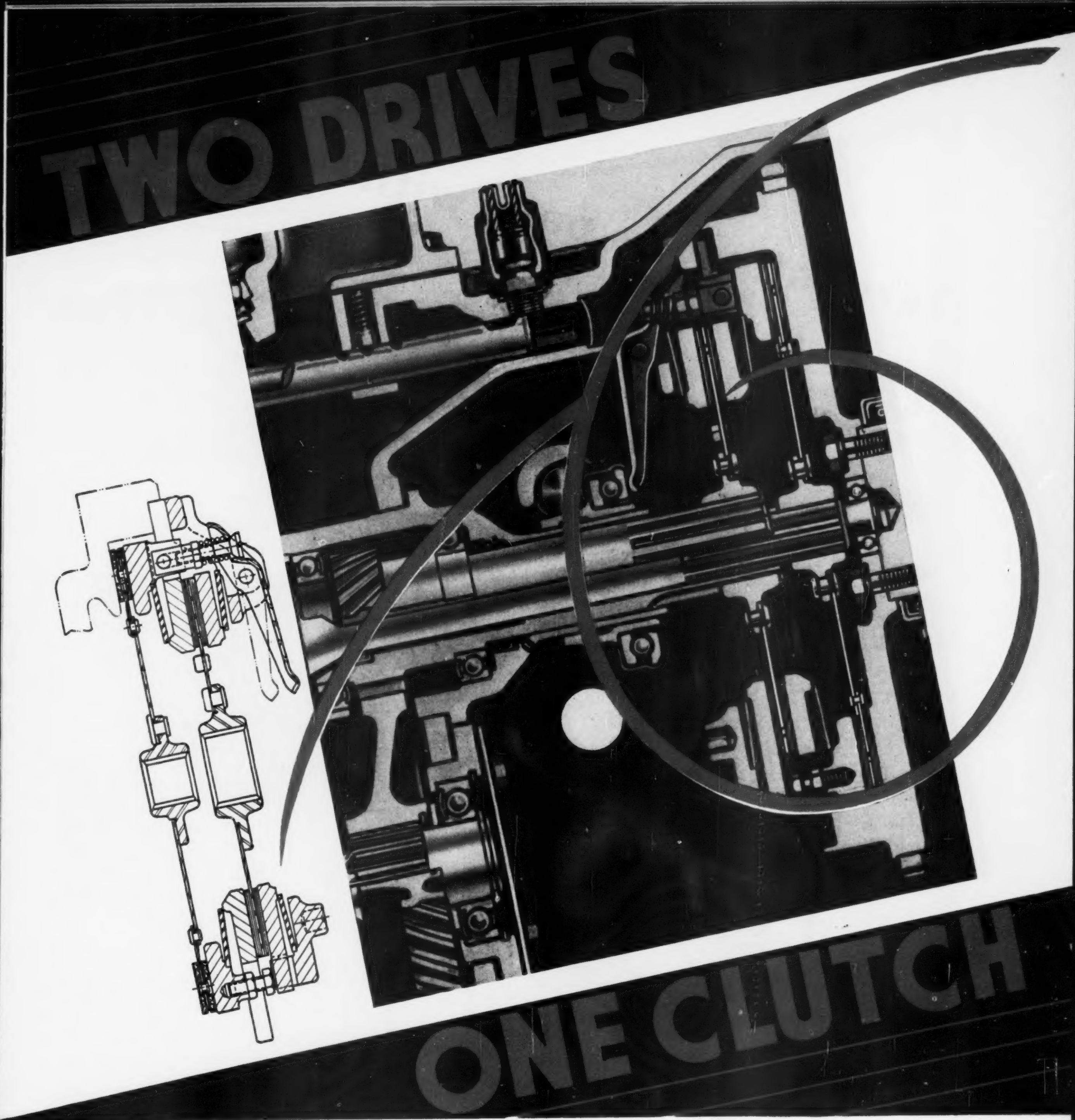
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ONE CLUTCH

Darcast keeps pace with Progress

The Control and Technique
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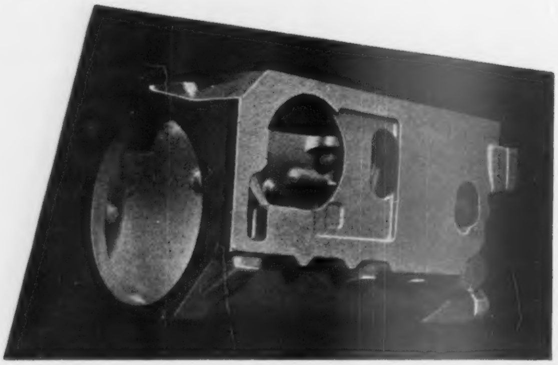
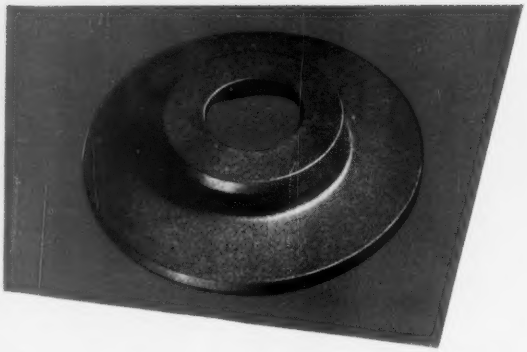
High Duty Iron Castings
by the Darcast Foundries
is ready to meet developments
of the Motor Industry.



AUTOMATIC GEARBOX CASE *(COURTESY OF ROLLS-ROYCE LTD)*



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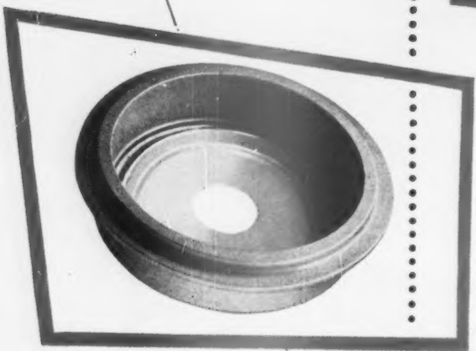

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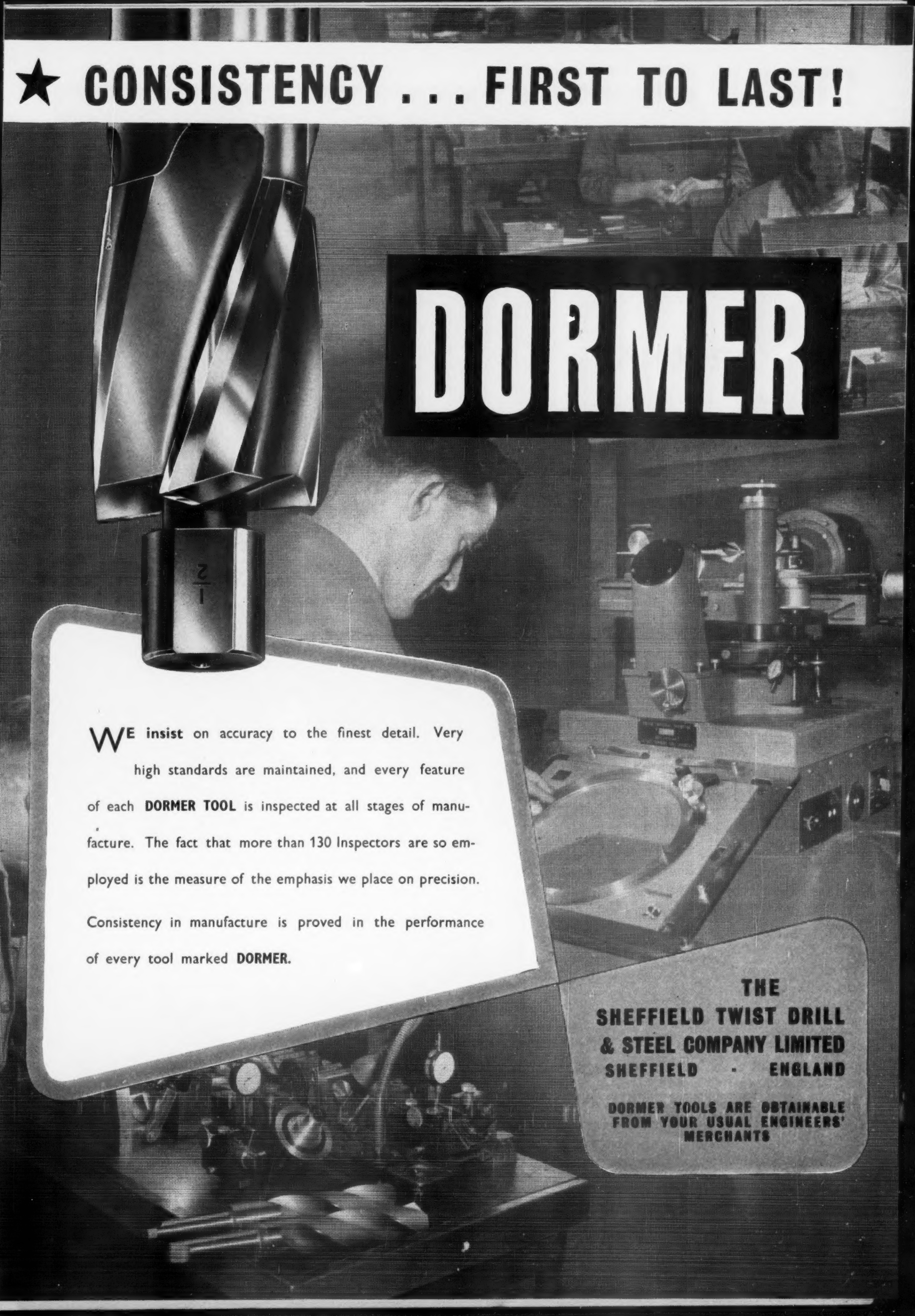
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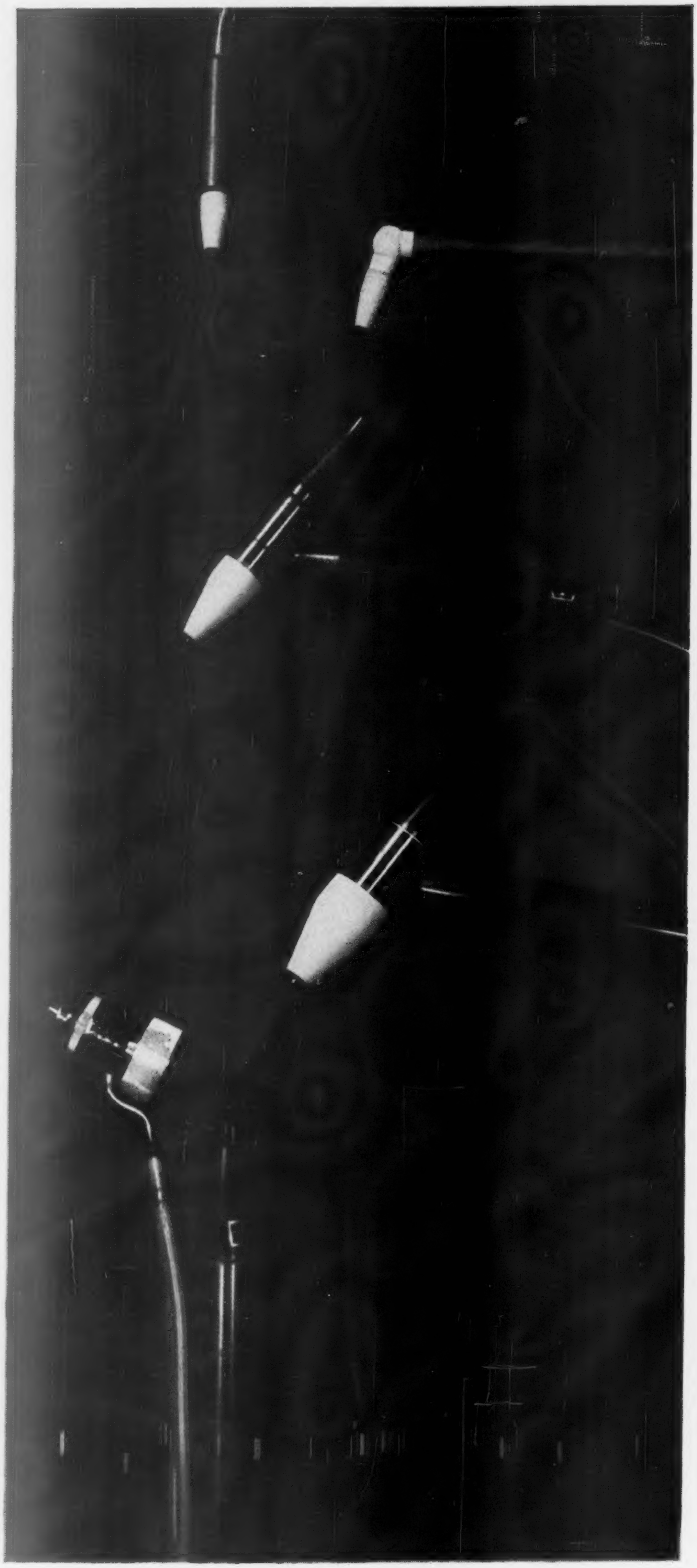
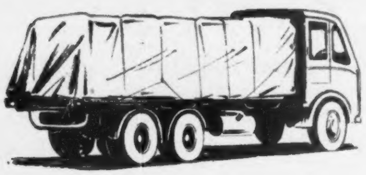
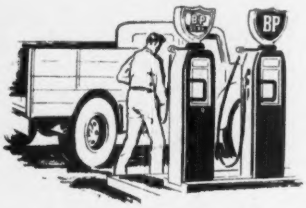




Attention all fleet owners



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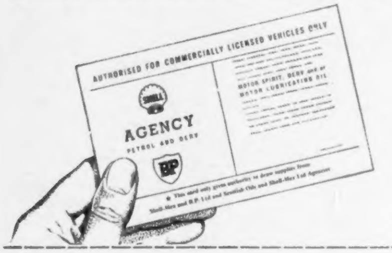

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wherever they are, they can pick up
supplies on credit or for
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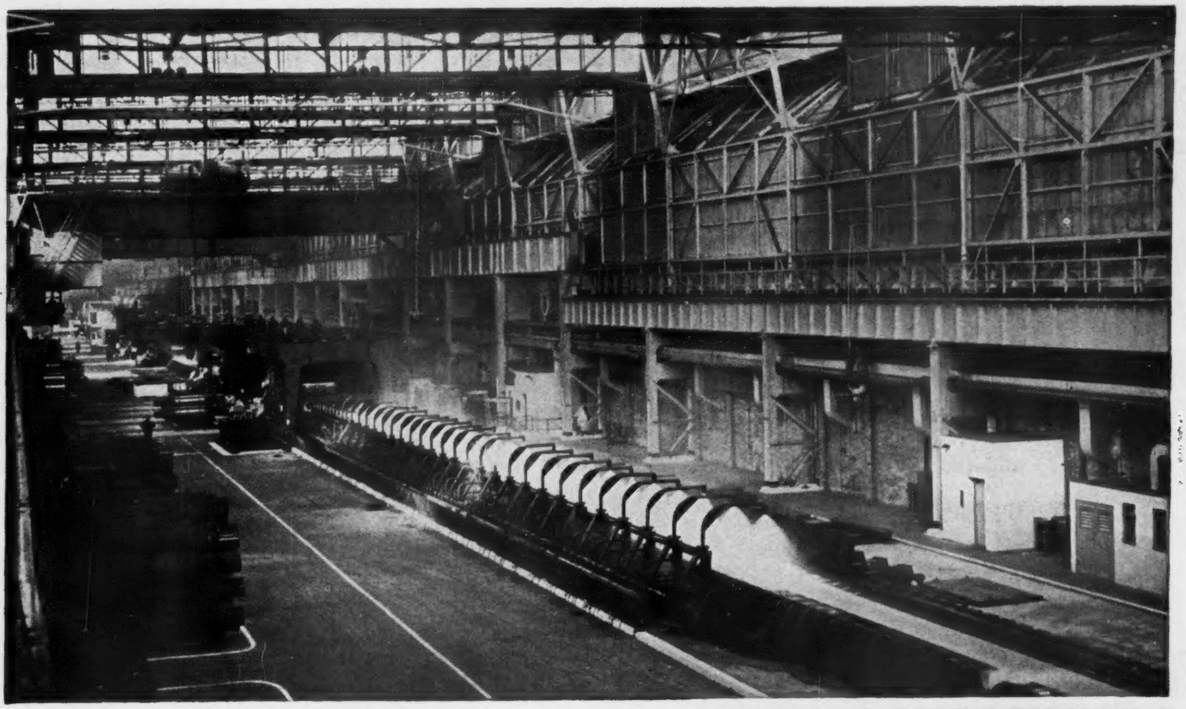

The 80″ continuous strip mill at Abbey Works

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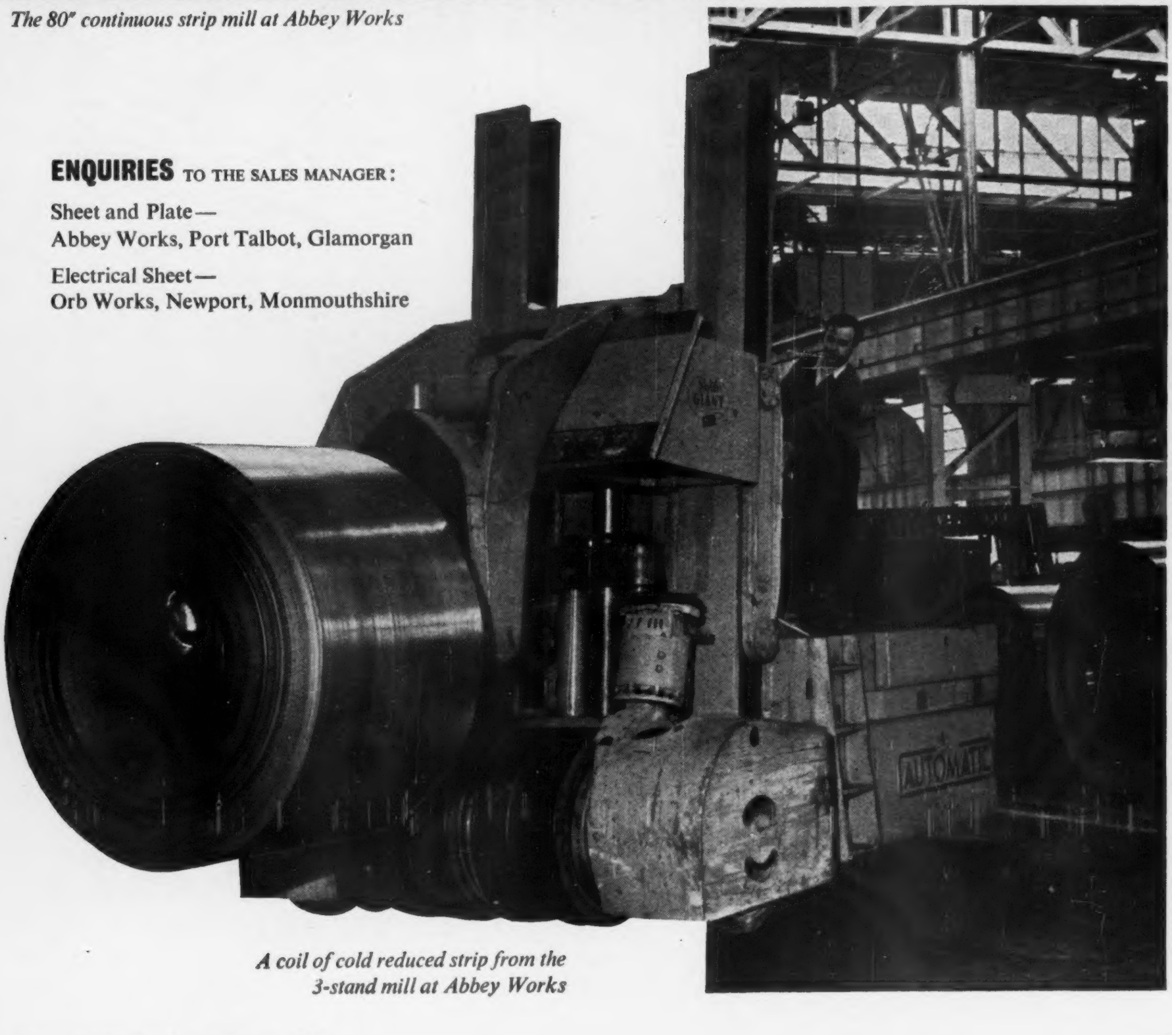
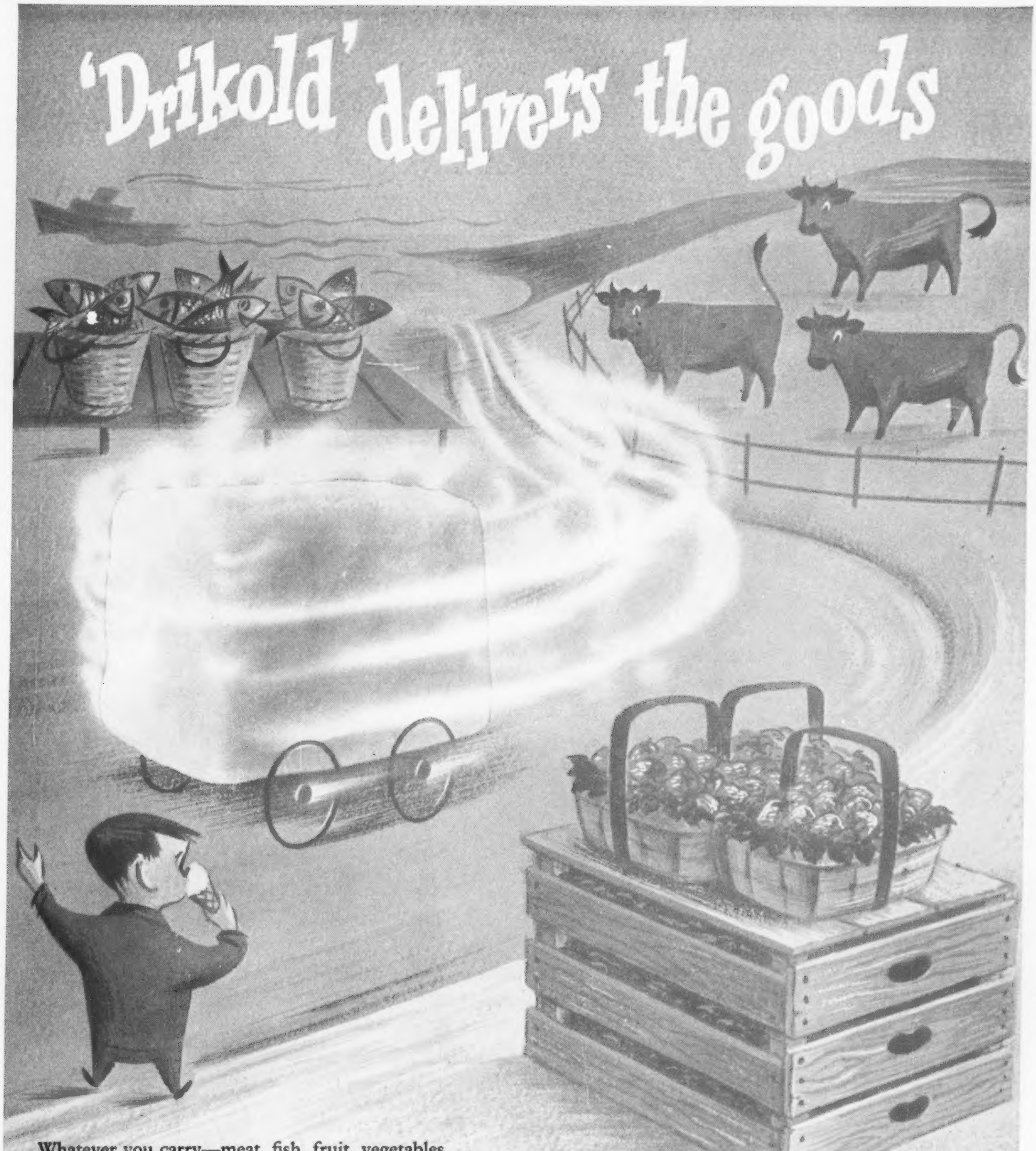


A coil of cold reduced strip from the 3-stand mill at Abbey Works


'Drikold' delivers the goods
Whatever you carry—meat, fish, fruit, vegetables, or ice-cream—'Drikold' dry ice provides the most efficient form of refrigeration for motor transport. Space-saving and labour-saving, with no need for skilled maintenance, no moving parts to go wrong, 'Drikold' makes sure that all foodstuffs arrive fresh and perfectly preserved. Check up on all its good points before ordering or building refrigerated vehicles.
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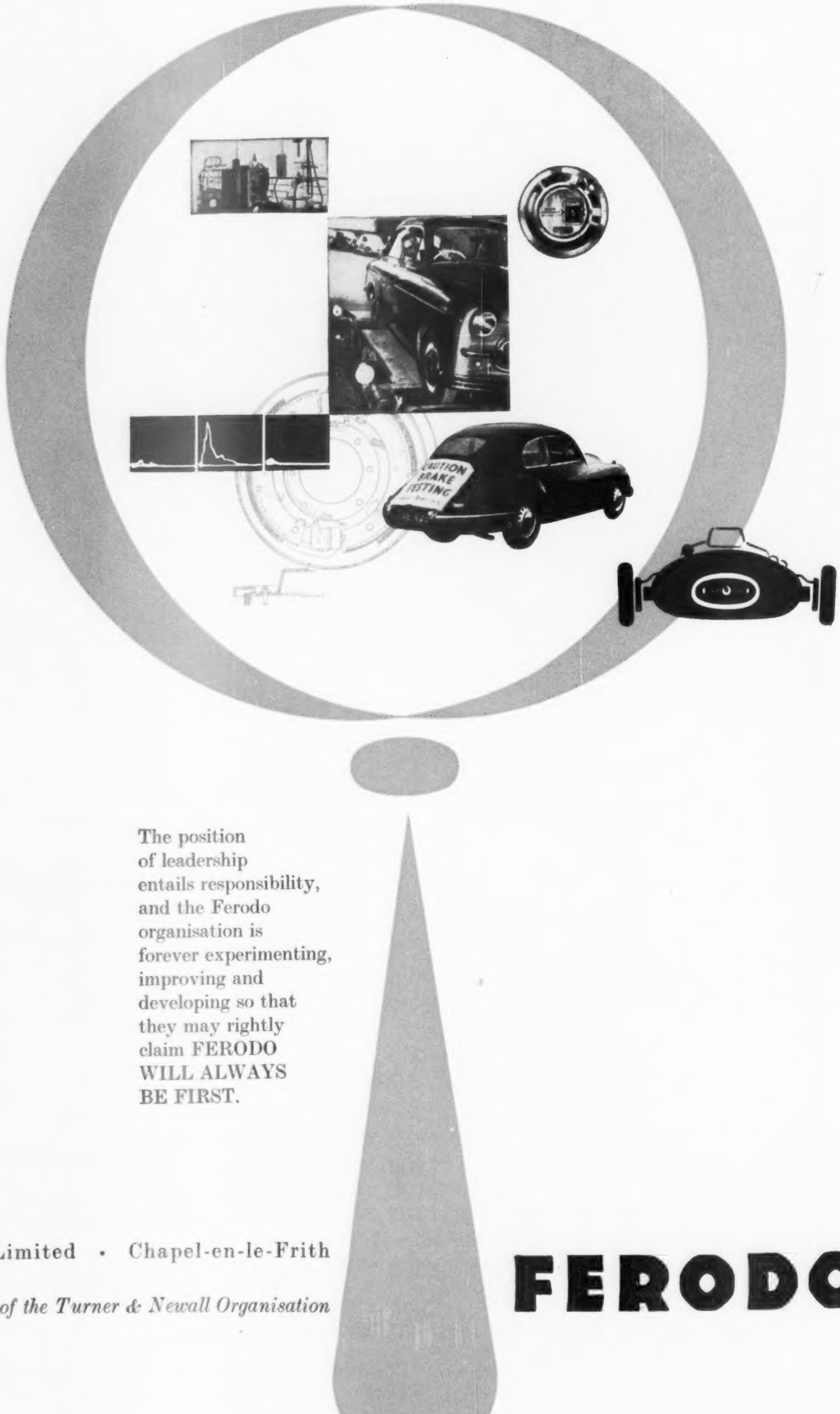
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The unrivalled experience and resources of the

'CASSEL' HEAT TREATMENT SERVICE

are at your disposal.

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. . . lies behind every recommendation made by the 'Cassel' Heat-treatment Service. A complete range of salts and salt-bath furnaces is manufactured to meet a wide variety of heat-treatment problems.

The 'Cassel' Process offers these advantages:

- *Simple, effective control of carburising and neutral baths*
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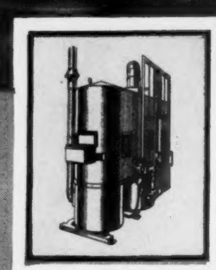

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Write for further details to:

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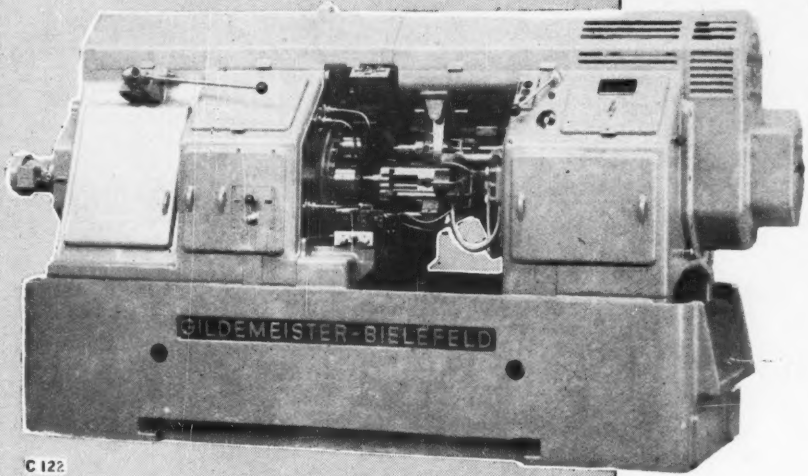
CC 180

GILDEMEISTER

multi-spindle automatics



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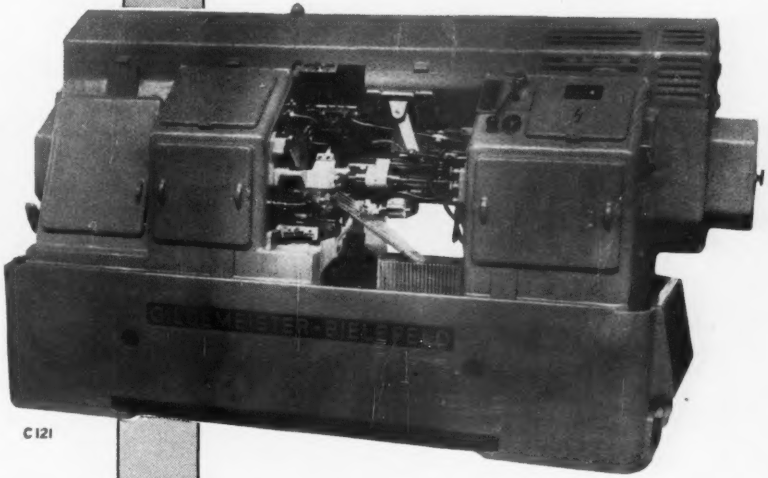
Renowned for their accuracy and output. They are reliable and efficient and permit easy operation and maintenance.

Special attachments are available for eccentric turning, slotting, spherical turning, copying, recessing, eccentric recessing, longitudinal turning, thread chasing and other operations.

The Multi-Spindle Bar and Work Rotating Chucking Machines are made in a range of sizes with either four or six spindles. The four-spindle bar and chucking machines have capacities up to $3\frac{1}{4}''$ dia. × $7\frac{7}{16}''$ long and $8\frac{1}{4}''$ dia. × $6\frac{7}{8}''$ long respectively. The six-spindle bar and chucking machines up to $2\frac{5}{8}''$ dia. × $7\frac{7}{16}''$ long and $7\frac{3}{8}''$ dia. × $6\frac{7}{8}''$ long respectively.

Tool Rotating Chucking Machines with maximum turning lengths of 6″ and $7\frac{1}{2}''$ are also available.



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Windscreen wiper motor and other accessories — secured by Avlok Anchor Nuts

Speed assembly... slash production costs

WITH AVLOK *Blind* ANCHOR NUTS ...

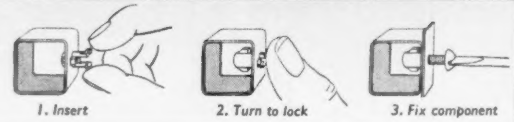
Avlok Anchor Nuts for securing wings

No wonder Avlok Anchor Nuts were chosen by these famous European car manufacturers for their latest models! They cut production time, reduce handling costs and completely eliminate the need for welded cage nuts, threaded inserts, screws and sheet tapping. Avlok Anchor Nuts not only bolt components together where only one side of the job is accessible—they can be easily detached and replaced whenever required. The spring-washer effect of the cage ensures that, when tightened, Avlok Anchor Nuts are perfectly shakeproof.

In the automobile industry, more and more use is being made of this revolutionary fastening system and we will be pleased to send you full details on request.

Just look at the advantages:

- *Rapidly fixed from one side*
- *Eliminates lockwashers*
- *Simply replaced from one side*
- *Absolutely shakeproof*
- *No welding or riveting*
- *Blind fixing of enormous advantage to designers*



AVLOK Regd. **ANCHOR NUTS**

AVIATION DEVELOPMENTS LTD., 229-231, High Holborn, London, W.C.1.

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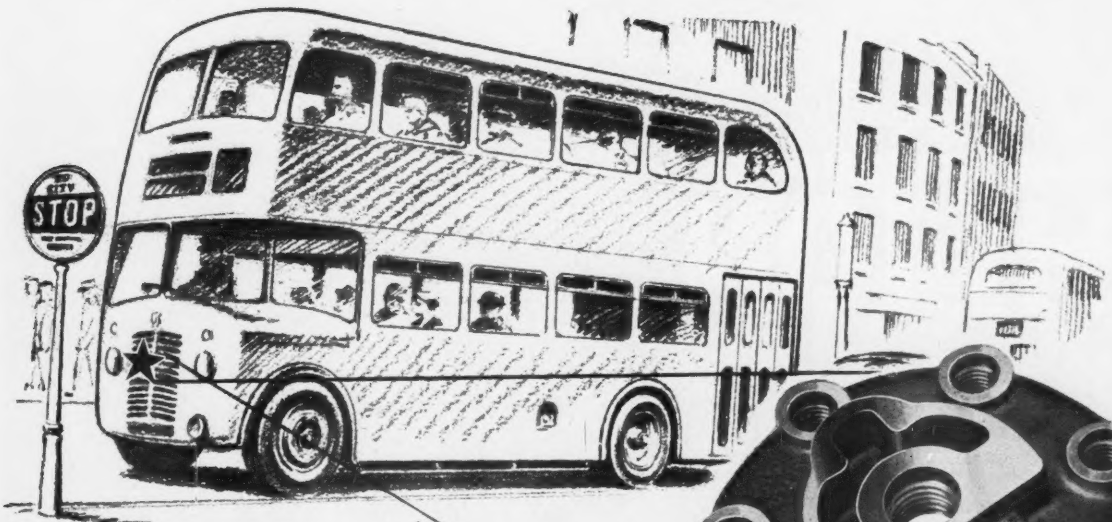
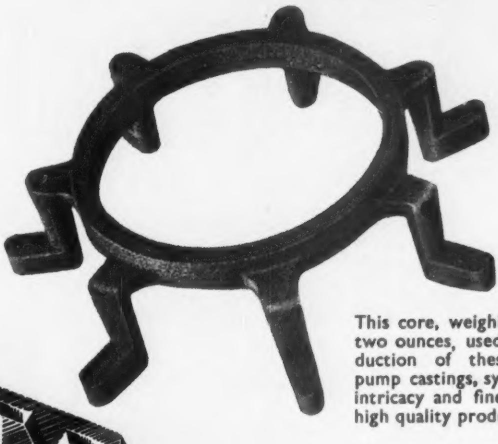


WHEN *SAFETY* MEANS SO MUCH
P·B·M CASTINGS ARE USED . . .

No vehicles make more constant use of braking power than Public Transport. That's why, when constant reliability is so vital, P·B·M Aluminium castings are specified by Lockheed for the hydraulic pump on their famous power-operated brakes in use on the buses of Midland Red Motor Services—one of the largest services in the country.



This core, weighing less than two ounces, used in the production of these hydraulic pump castings, symbolises the intricacy and fineness of this high quality production.

P·B·M **GRAVITY DIECASTINGS**

MACHINED CASTINGS & ASSEMBLIES . . .

P·B·M have excellent facilities for supplying castings as fully machined and assembled components.

PERRY BARR METAL COMPANY LTD., GREAT BARR, BIRMINGHAM 22A

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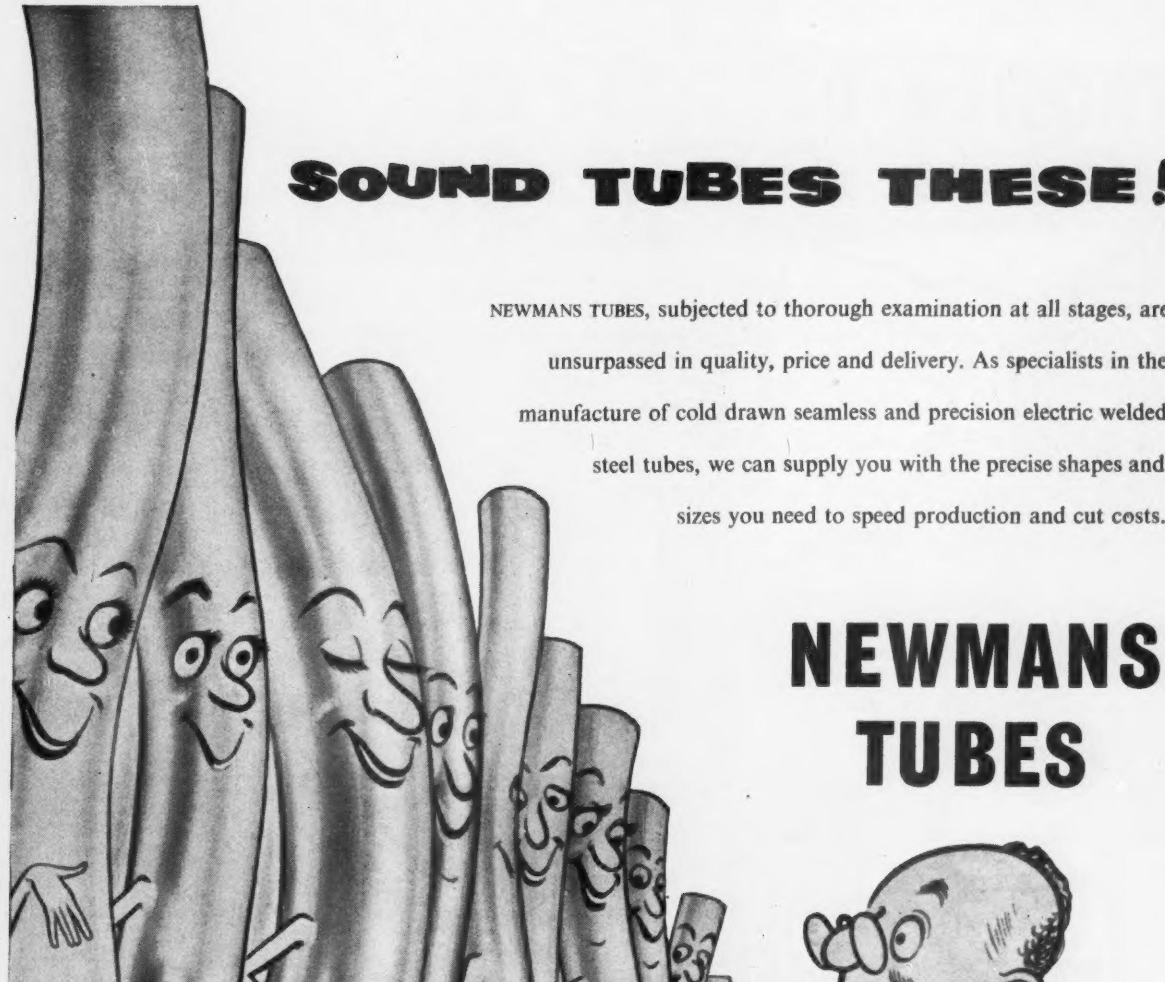

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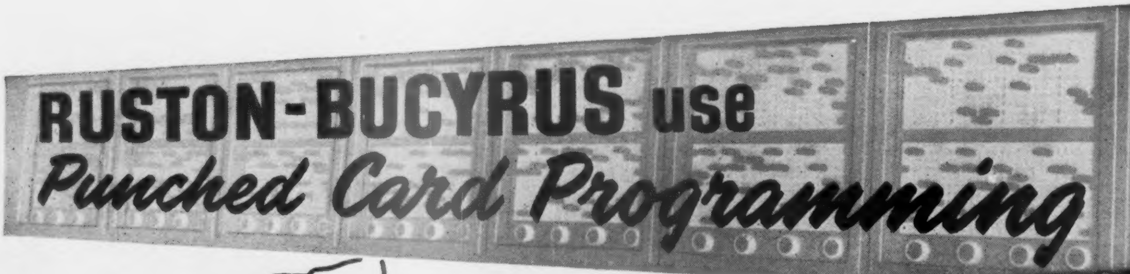
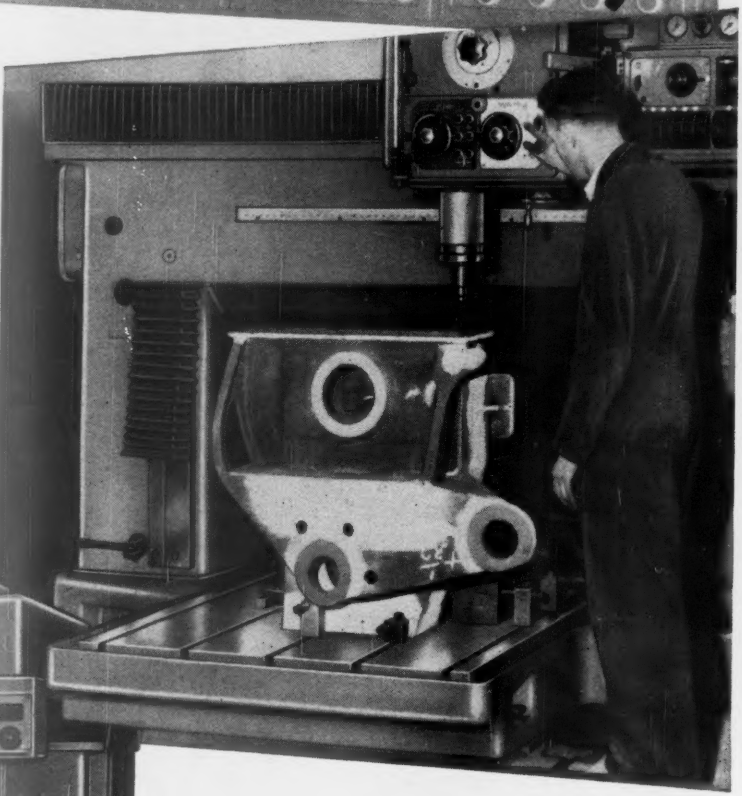
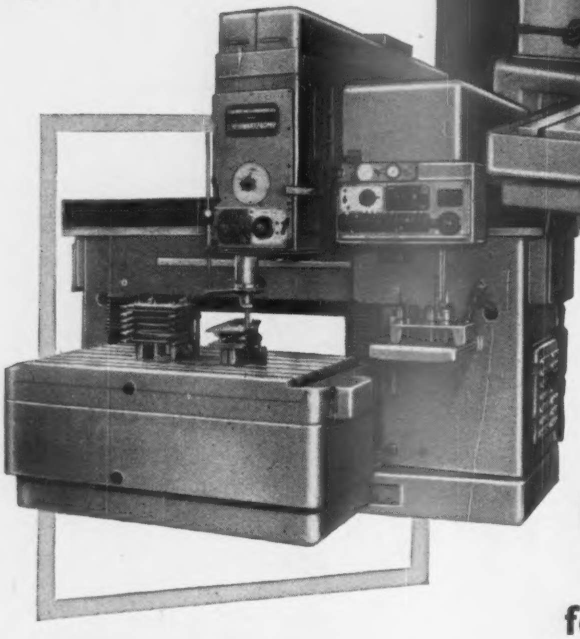

RUSTON-BUCYRUS use
Punched Card Programming



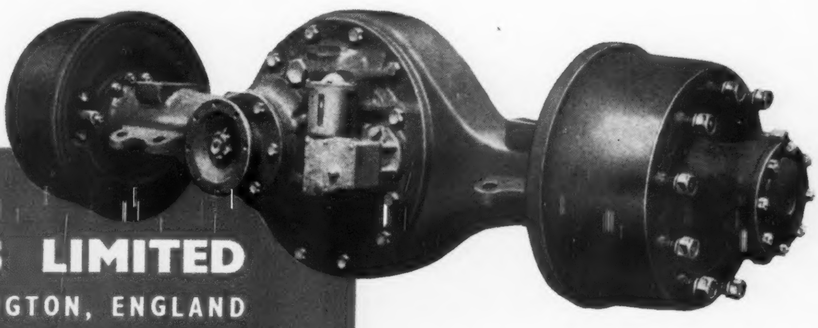


TIME LOST IN HOLD-UPS EASILY MADE UP WITH

EATON 2-Speed AXLES

With an Eaton 2-speed Axle it's easy to make up time lost in traffic hold-ups! First there's the improved acceleration you get because the doubled range of gears lets you select just the right gear, for once on the open road you go bowling along on an extra high gear without tearing the guts out of your engine. And of course you save all the time on fuel and engine wear, to say nothing of the comfort and ease of mind which comes from always being on top of the job.

EATON
2-SPEED AXLES



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VICTORIA ROAD, WARRINGTON, ENGLAND

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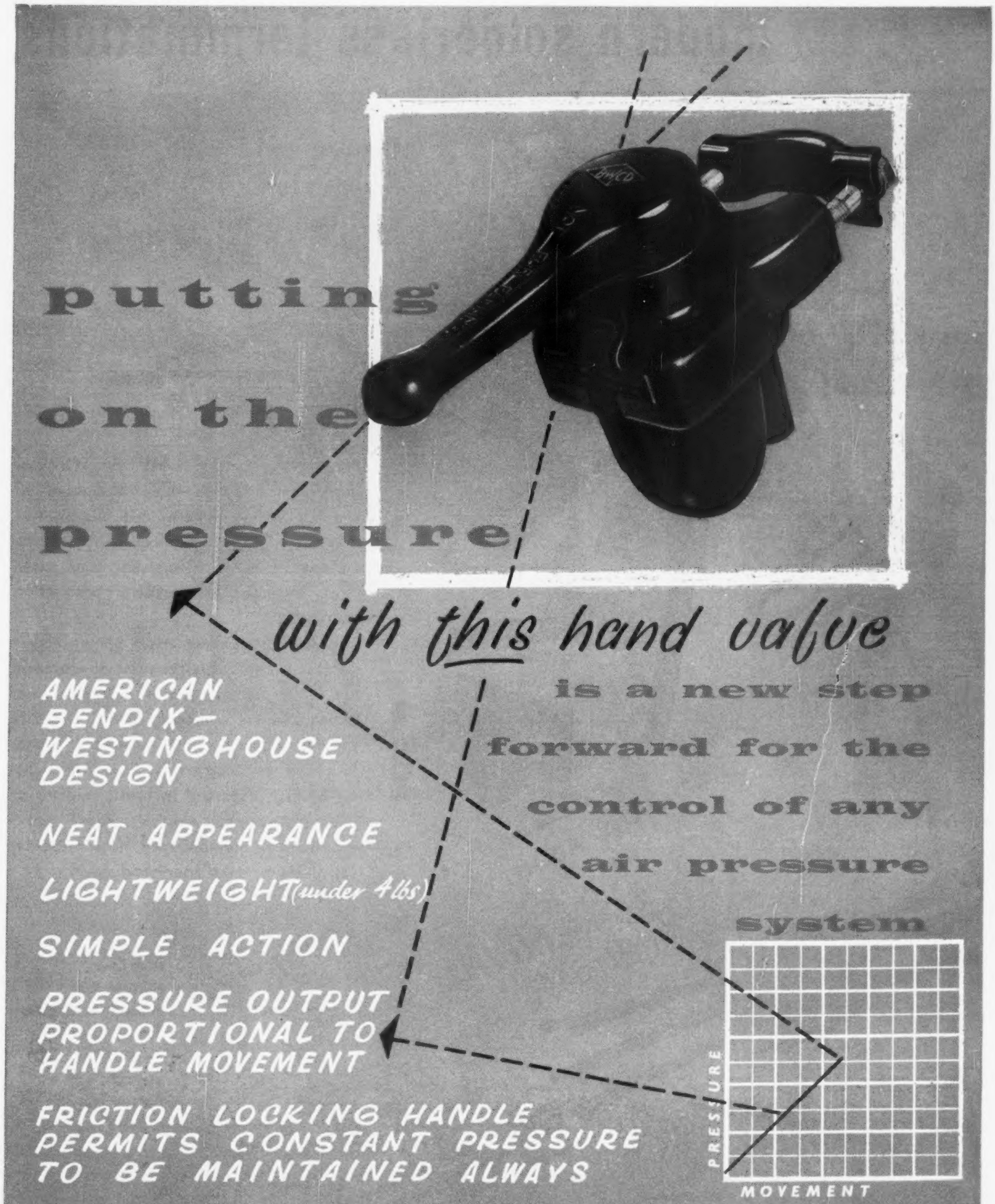

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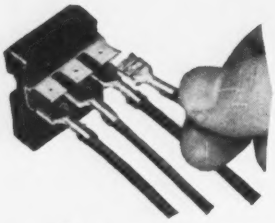

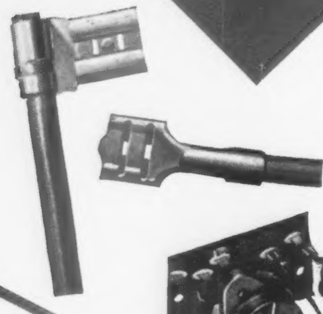

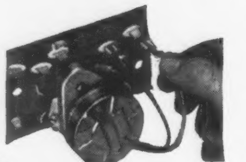

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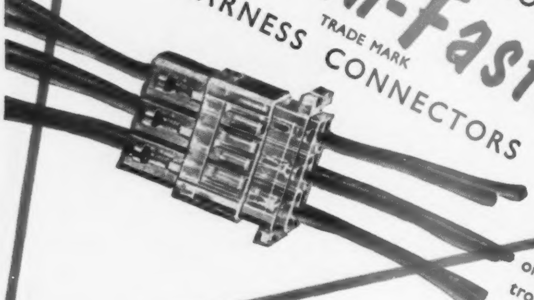

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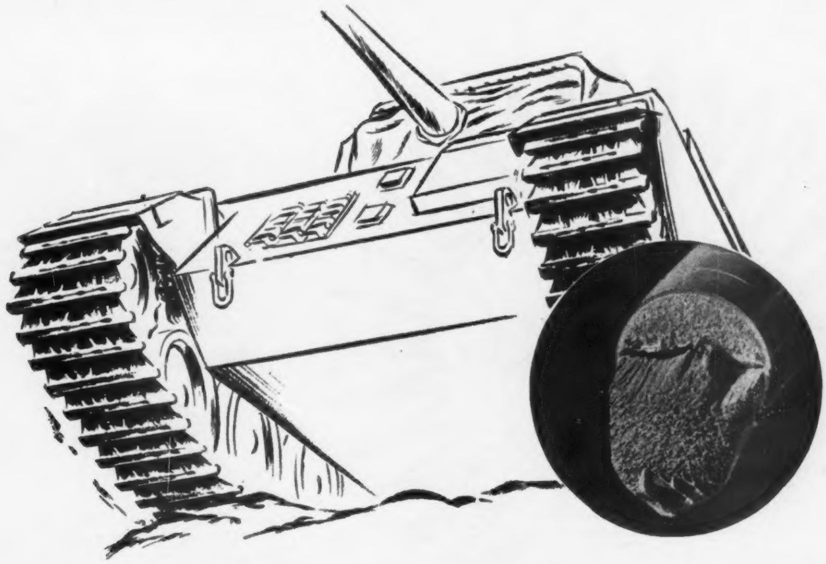

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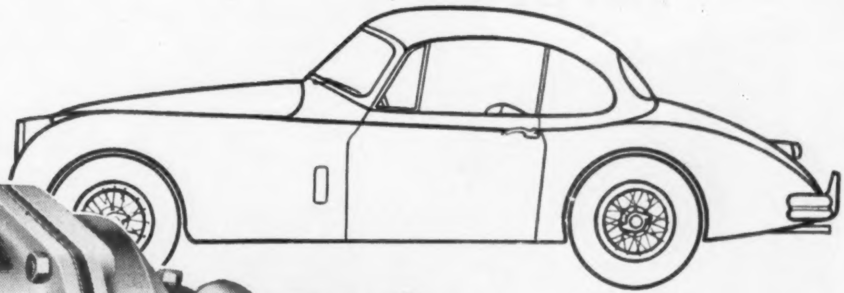

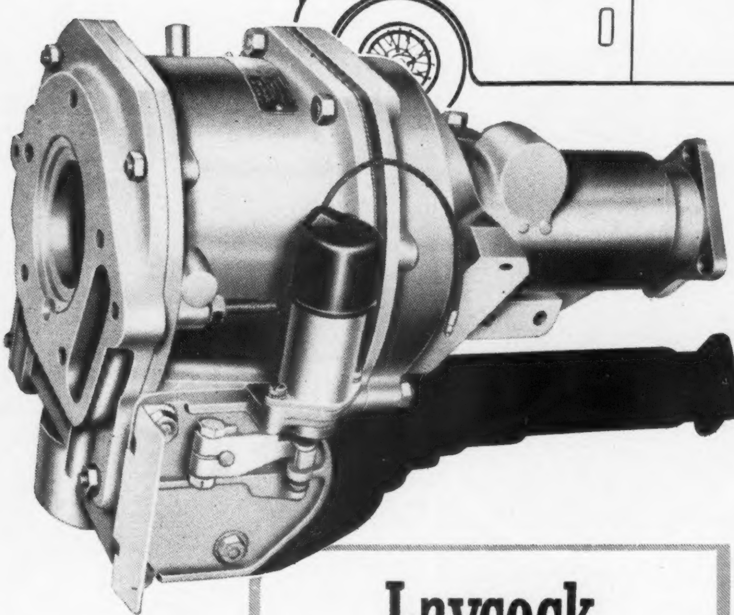

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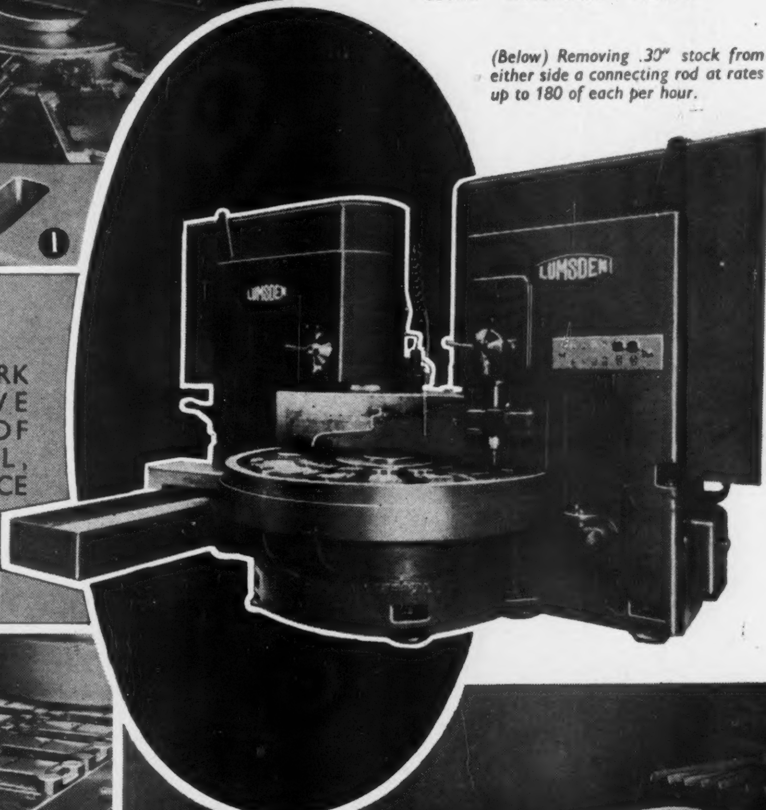

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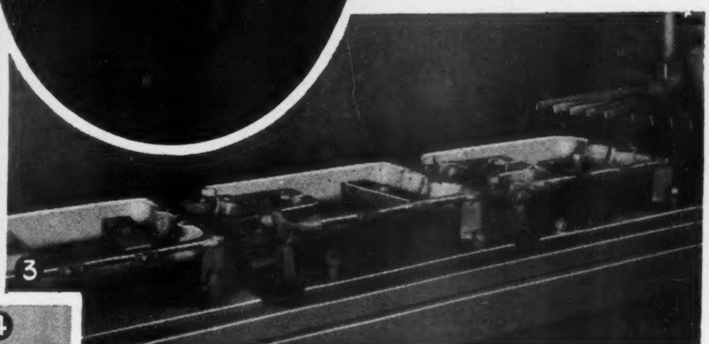

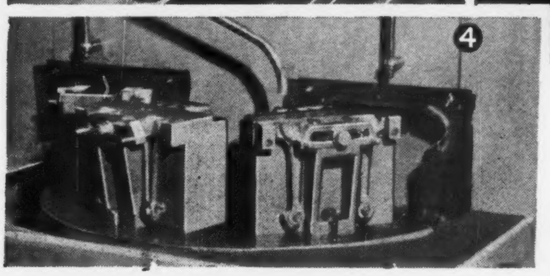

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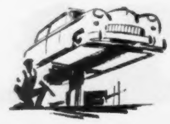

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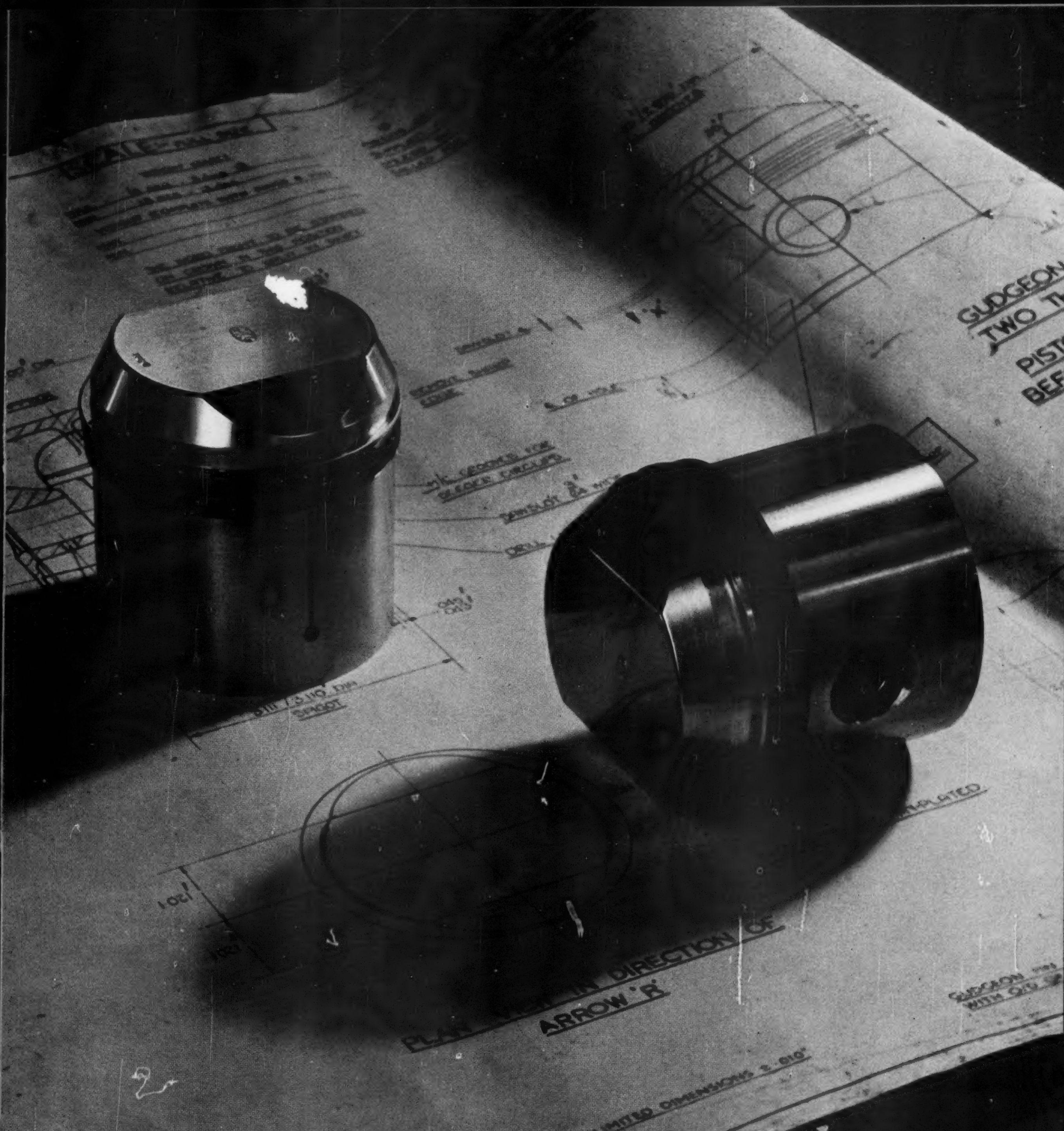
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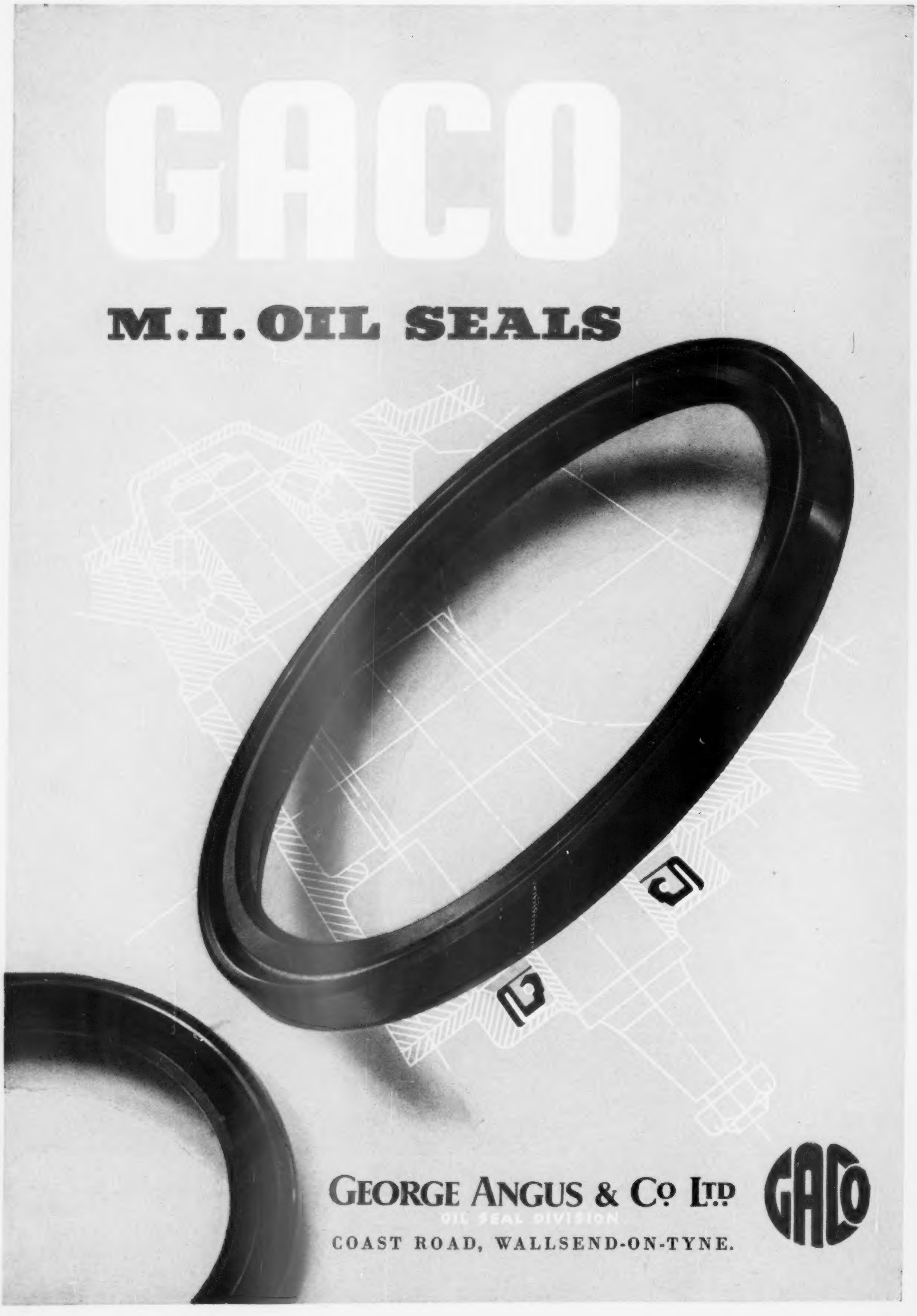
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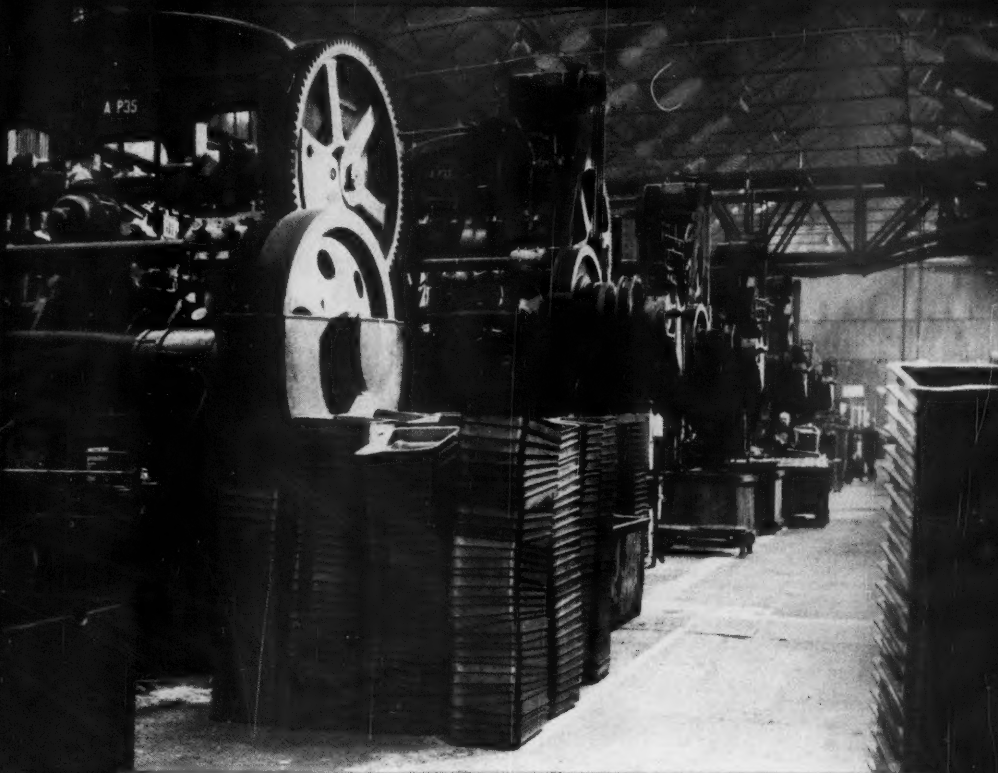

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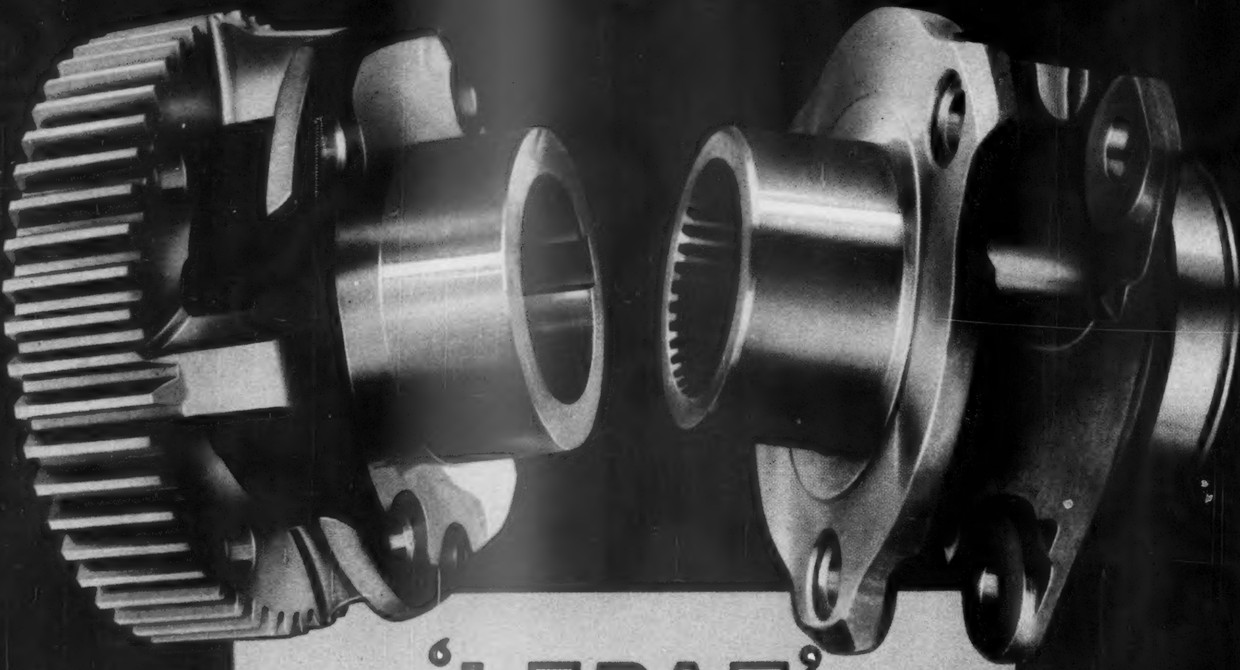

AUTOMOBILE ENGINEER

CONTENTS

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VOLUME 47 **NUMBER 8** **AUGUST 1957**

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Personnel and Automatic Production

CONGRATULATIONS are due to the Institution of Production Engineers for the manner in which the "Automatic Production—Change and Control" Conference was organized and for the matter that was presented. The time given to the consideration of the human problems involved was particularly welcome. That the increasing mechanization of industry brings human problems in its train cannot be doubted, and although it is as yet too early to estimate confidently what changes will take place, some of the papers presented did at least indicate probable, or desirable developments.

Technological advances at the quickest possible rate are of vital importance to this country, but if they are to be effectively applied, new concepts of management must be developed. In this connection, two papers presented at the Conference were of particular interest; "Control and Communications—a Management Concept of Cybernetics" by Miss J. Woodward and "The Analysis of Human Behaviour" by Professor R. W. Revans.

Miss Woodward's paper discussed the relationship between technical change and management structure, and was based on a research project that surveyed 203 firms in East London and Essex. Essentially, the firms studied can be classified in three main groups: unique article and small batch production; large batch and mass production; and process production. Comparison between the groups gives some indication of the effects of automation.

Certain of the information obtained suggested that technical demands influenced not only the organizational structure, but also the general atmosphere and the quality of human relations in a factory. Miss Woodward says: "Pressure on people at all levels of the industrial hierarchy appears to increase as technology advances, until it reaches its maximum in mass production of the assembly line type. It then decreases rapidly."

Two reasons are advanced for the rapid decrease of pressure from the mass production to the process industries; the labour force is broken down into very much smaller groups, and the process firm is less concerned with labour economy because of its comparatively low labour costs. These two factors make the industrial relations situation very much easier generally. From this the conclusion is drawn that automation and other forms of advanced technology may in the long run solve many of our current industrial relations problems, and what is equally important, they may relieve both operators and managers alike from the pressure that inevitably arises from mass production of the older type, and from the tensions in inter-personal relationships which are inevitably associated with it.

The above remarks are, of course, apposite to the automobile industry, since it is *par excellence* the mass production industry of the assembly line type. This is not to suggest that management in the automobile industry is worse than in any other industries, but the events of the past two or three years do suggest management-labour relations have not kept pace with technological advance.

Some tentative conclusions as to the reasons why management-labour relations may be difficult in the automobile industry may be drawn from Professor Revans' paper on "The Analysis of Human Behaviour". A study of statistics, particularly of the mining industry, shows that there is strong evidence that the larger the establishment in terms of numbers employed, the more trouble there is with strikes, absenteeism and accidents. This, Professor Revans thinks, is because of the increasing difficulty in communication between management and workers as size increases.

As Professor Revans points out, the securing of good relations with labour is a management function. He says: "It is the task of management to see that its authority is distributed in such a way as to give all employees, both workers and foremen, the level of problems that they can settle by taking thought or by consulting other people; it is to find, and to use, the most effective distribution of authority that the methods of operational research must be employed. Management, in other words, must study not only systems of technology, of selling, and so forth; it must study the articulation of them all into the business of management. It must, above all, not be afraid to give authority to its subordinates."

Technically, the automobile industry has, since the war, been the most forward looking of the engineering industries of this country. In the production field, new methods and techniques have been investigated, developed and adopted in an amazing degree, but the concentration on technological problems tended to allow the human problems involved to be neglected, if not completely disregarded. This, we are happy to say, is no longer completely true of the automobile industry, and it is to be hoped that the fact that papers such as those of Miss Woodward and Professor Revans were presented at a conference of production engineers is evidence that even greater attention is now to be paid to the human problems.

GIRLING VACUUM-SERVO BRAKE CONTROL

A Unit which, if Properly Applied, Obviates Brake Fade Difficulties and Reduces the Effort Required of the Driver

CONSIDERABLE operational experience has now been obtained with the Girling AHV689 vacuum-servo unit, illustrated in Fig. 1. This experience has proved conclusively not only that the designers have satisfactorily attained their primary object, which was to reduce the pedal effort required of the driver, but also that the system can be operated for periods of as long as half the life of the vehicle without any maintenance other than cleaning the air filter. When the unit is used in conjunction with the two-trailing-shoe type of brake layout, the adverse effects of brake fade due to varying frictional characteristics of the shoes are minimized. Another advantage of the unit is that it enables the pedal load and travel to be reduced.

One of the first questions that arises is why is it now desirable to use a servo unit in types of vehicles that apparently have previously been operated satisfactorily without one? The answer to this is that in recent years, vehicle weights and performances have tended to increase; traffic conditions in this country have become much more congested; and British vehicles are now operating in all parts of the world, where both terrain and climatic conditions are much more severe than in Great Britain: as a result, brakes are more heavily loaded than ever before and fade resistance is a most important requirement. Moreover, it is only by the employment of a servo unit that vehicle manufacturers are able to conform fully with the modern trend towards the reduction of pedal travel and operating load.

Shoe arrangement

The need for vacuum-servo assistance and the performance of the brakes are both largely dependent on the shoe arrangement. Undoubtedly, the simplest design is the leading-and-trailing-shoe brake. Since brakes of this layout are equally effective whether the vehicle is going forwards or in reverse, they are particularly suitable for the rear wheels, because there is no danger that the hand brake will fail to hold when, for example, the vehicle is facing uphill. However, they require appreciably greater pedal effort than brakes of the two-leading-shoe type of layout.

In recent years, as vehicle weights and speeds have increased, the two-leading-shoe type have become increasingly popular for application to the front brakes. Unfortunately, this layout is more prone to fade than most, and therefore is not always entirely satisfactory under the most arduous conditions of operation. The best arrangement, so far as fade resistance is concerned, is the two-trailing-shoe layout, which, therefore, is most suitable for application to the front wheels of cars because these wheels absorb from 60 to 70 per cent of the total amount of energy required to be dissipated during the braking of the vehicle. Since a greater pedal force is required for the actuation of the two-trailing-shoe type of brake than for any other, it is desirable to use a vacuum servo in conjunction with this layout.

Theory

Since the reason for the good fade resistance of the two-trailing-shoe type arrangement is not well known, a brief summary of the theory will be of interest. Basically, fade is due to reduction in the coefficient of friction of the lining material as its temperature increases. It does not generally occur noticeably during normal operation of the brakes on the popular light and medium types of car; but in the heavier and faster cars, particularly when used under conditions in which the brake applications are of long duration or are made at high frequency, fade can be serious because much higher temperatures are attained.

Fig. 2 illustrates diagrammatically the forces acting on a leading shoe and on a trailing shoe. These diagrams are of a typical brake layout, with a 10 in diameter drum and a lining of such a length that the angle subtended between the lines joining its ends to the centre of the brake assembly is 90 deg. The lining is equally disposed about the horizontal line joining its centre of pressure and the centre of the brake assembly. The length and disposition of the lining gives a virtual centre of drag at a point 5·5 in from the centre of the drum. If the coefficient of friction μ is 0·4, a nominal force of 100 lb between the lining and the drum will produce a braking effort, or drag, of 40 lb acting at this radius of 5·5 in. Then if, because of fade, the coefficient of friction is reduced to 0·2, the normal force required to maintain the same drag force of 40 lb is 200 lb. From the diagram, it can be seen that with a leading shoe, the resultant of the normal force and the drag is 108 lb when μ is 0·4, and 204 lb when μ is 0·2.

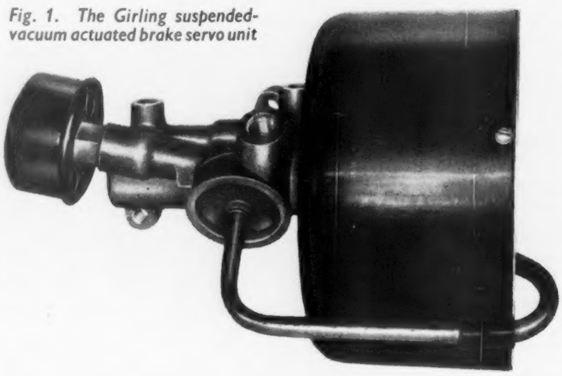


Fig. 1. The Girling suspended-vacuum actuated brake servo unit

The important feature is the difference between the couples of these two forces about the shoe pivot: when $\mu = 0{\cdot}4$, the length of the effective lever arm about the pivot is 1·75 in, whereas when $\mu = 0{\cdot}2$ it is 2·94 in. It follows that if $\mu = 0{\cdot}4$, the shoe tip force required to give a braking force, or drag, of 40 lb is:

$$\frac{108 \times 1{\cdot}75}{8} = 23{\cdot}6 \text{ lb}$$

and if $\mu = 0{\cdot}2$, it is:

$$\frac{204 \times 2{\cdot}94}{8} = 74{\cdot}6 \text{ lb}$$

Thus, the additional input force required if the effectiveness of the brakes is to remain unaltered after this degree of fade

has occurred is 215 per cent. In the case of the trailing shoe, when $\mu=0{\cdot}4$, the shoe tip force is:

$$\frac{108\times5{\cdot}75}{8}=77{\cdot}7\text{ lb}$$

When $\mu=0{\cdot}2$, it is:

$$\frac{204\times5}{8}=127\text{ lb;}$$

this represents an additional input force of only 156 per cent.

It can be seen that the reason why fade does not so greatly increase the effort required to actuate a trailing shoe, by comparison with that required for a leading shoe, is that in the former case fade decreases the effective lever arm of the resultant force about the shoe pivot, whereas with a leading shoe the advent of fade increases length of the effective lever arm.

General practice

In general, a brake of the two-trailing-shoe layout has a shoe factor—which is defined as the shoe drag divided by shoe tip force—of about one third that of an equivalent size of two-leading-shoe brake, For applications in which the performance of a two-trailing-shoe type brake is required to be equal to that of a two-leading-shoe type of the same size a vacuum servo unit must be employed.

At least two manufacturers have adopted brake systems in which all the braking effort is applied by the servo unit. These systems really should be described as power braking, as opposed to the more widely favoured power-assisted braking arrangements. The reason why power-assisted braking is more popular is that in the event of failure of the power system, the brakes can still be applied directly by the driver. When power braking systems are employed, it is usual to employ separate hydraulic circuits to the front and rear wheels, so that a pipe fracture in one part of the system will not cause total failure of the brakes: partial failure serves as a warning to the driver that something is wrong and that the vehicle requires servicing attention. The Girling AHV689 vacuum-servo unit is designed to give power assistance, but the situation is constantly under review and, should the demand arise, no doubt a unit capable of giving full power braking would be developed.

Girling vacuum-servo unit

This unit is incorporated in a conventional hydraulic braking system. It is introduced between the master cylinder and the wheel cylinders: that is, the outlet pipe from the master cylinder is connected to the hydraulic inlet of the servo unit, and the hydraulic outlet of the servo unit is connected to the brake cylinder. In the servo, a relatively large diameter piston, actuated by the pressure differential between atmosphere and the manifold depression, is connected by a rod to a smaller diameter piston, which boosts the pressure in the hydraulic circuit to the wheels. The air pressure differential generally is in the order of 10 lb/in^2. When the brake is not in operation, both sides of the vacuum piston are subjected to manifold depression: thus, the arrangement is of the type generally described as the suspended-vacuum system.

The reason why induction manifold depression is employed to provide the servo force is that it is at present the most convenient source of power on petrol engines. The only other readily available source is the hydraulic pressure output from the engine lubrication pump. This, of course, can hardly be regarded as satisfactory for application to external systems because of the danger that the engine would be severely damaged in the event of loss of lubricating oil pressure owing to a pipe fracture. Should other sources of power become available in future, for example, if air suspension were employed and a compressor installed, or if a hydraulic system were to be used, as on the Citroën DS19, the situation would be reviewed and possibly the unit redesigned to make use of the alternative source of power.

The advantages of the suspended vacuum arrangement, as compared with the direct vacuum system, are as follows. The demand on the source of vacuum immediately the brakes are applied is less, and it follows that the unit tends to be more rapid in response to the control. Only one side of the piston has to be connected to atmosphere, and therefore, the risk that the mechanism may be damaged by the entry of foreign matter is reduced. The valve arrangement tends to be simpler because there are only two ports, one for the connection to atmosphere and the other for the connection to the source of vacuum.

Features of outstanding interest

A noteworthy feature of the Girling system in particular is that the servo cuts in when the hydraulic pressure is about 16 lb/in^2. This corresponds to about 2 lb pedal pressure; therefore, the cut-in is not felt by the driver, and braking is progressive throughout the normal range of pedal travel. The response and output of the servo unit is independent of friction, except that of the hydraulic piston that actuates the valve. Another advantage of the unit is that the servo assistance is limited by the degree of vacuum and the area of the servo piston, so the hydraulic system cannot be overstressed. If the pedal is moved beyond the point at which maximum servo assistance is developed, the additional pressure in the hydraulic system is limited, in the normal manner, by the maximum pressure that the driver is physically capable of exerting.

Among the other noteworthy features that will become apparent as the design is discussed in more detail is the care that has been devoted to anti-corrosive treatment of all components. This undoubtedly is a contributory factor to the long trouble-free life obtained. Also, ingenuity has

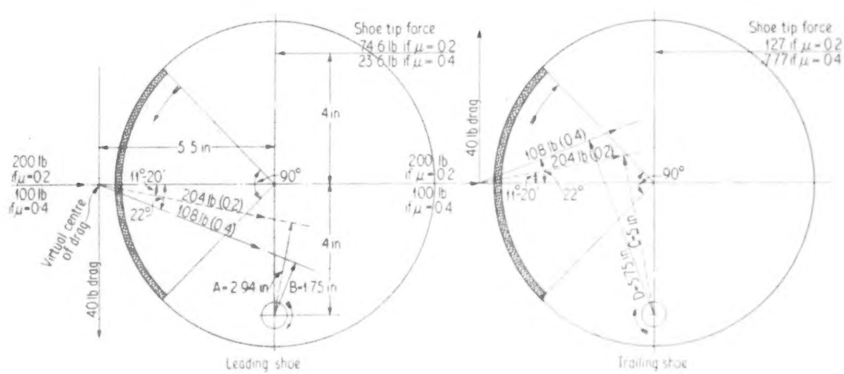
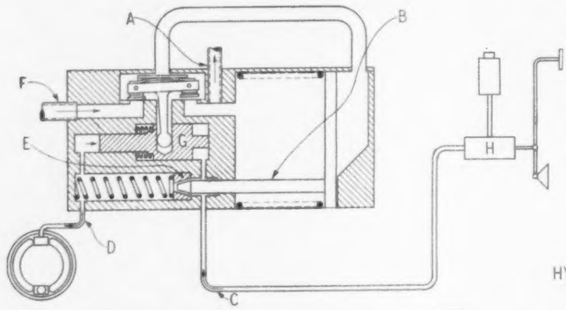


Fig. 2. Diagrammatic illustration of the forces acting on a leading and on a trailing shoe. This shows that with a trailing shoe layout fade tends to decrease the length of the effective lever arm of the resultant force about the shoe pivot, whereas with a leading shoe fade tends to increase the effective length of this arm

been displayed in designing the unit in such a way that the possibility of entry of foreign matter into the air and hydraulic cylinders is minimized.

It is of interest to note that nylon has been employed for some of the components. Nylon has advantages which, for certain applications, more than offset its relatively high cost. For very small components, such as the valves in this servo unit, the nylon can be injection-moulded in one operation to



A the vacuum connection to induction manifold; B servo piston rod; C hydraulic input from master cylinder; D hydraulic output to brakes; E output piston and valve; F air inlet; G control piston

Fig. 3. Diagram showing the principle of operation of the brake servo unit

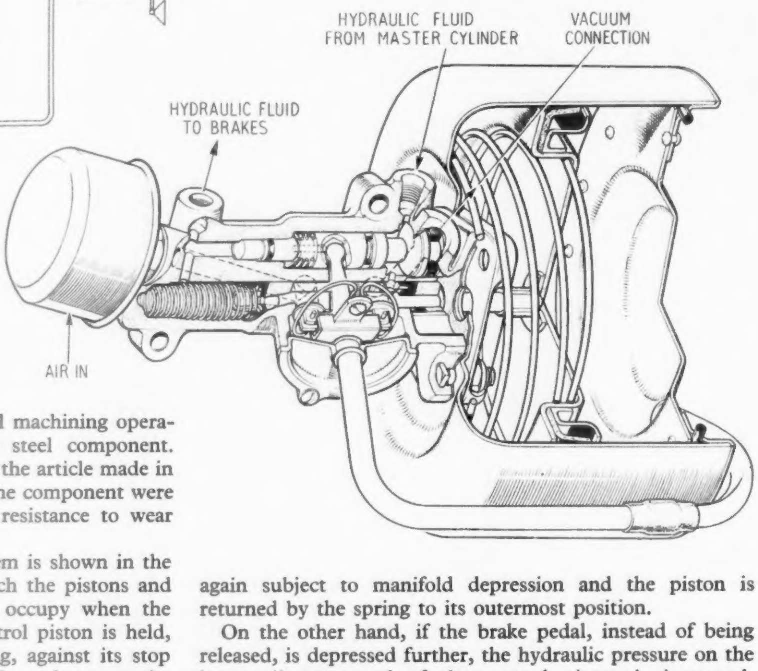


Fig. 4. Sketch of the servo unit, sectioned to show its layout and the general arrangement of the components

produce the finished part, whereas several machining operations might be required on a similar steel component. Therefore, in many instances, the cost of the article made in nylon is, in fact, less than it would be if the component were made of metal. Nylon also has a high resistance to wear and general deterioration.

The principle of operation of the system is shown in the diagrammatic illustration, Fig. 3, in which the pistons and valves are shown in the positions they occupy when the brakes are in the off position. The control piston is held, by the compression in a light coil spring, against its stop adjacent to the vacuum cylinder, and the valve over the vacuum port is lifted so that the servo piston is subject to induction manifold depression on both sides and is retained in its outermost position by another light coil spring. When the brake pedal is depressed, the hydraulic output from the master cylinder is transmitted to the servo unit. In this unit, the pressure is transmitted through the clearance between the tapered end of the servo piston rod and its seating in the output piston, and thence into the hydraulic circuit that serves the wheel brakes. At the same time, the hydraulic pressure is communicated to both ends of the control piston. Since one end of this piston is of larger diameter than the other, and the pressures are equal on both sides, this piston is moved against the compression in its return spring and rocks the T-shaped valve, thus closing the vacuum port and opening the air inlet port. Air then enters the valve chamber and passes through the pipe connection between this chamber and the outer end of the servo piston.

Under the influence of the pressure differential, the servo piston moves inwards against its light return spring, and the end of the piston rod is pushed home on to its conical seat on the output piston. Further movement of the servo piston rod is transmitted directly to the output piston and adds to the pressure already generated by the master cylinder. The extra pressure is communicated to the smaller end of the control piston; when this pressure balances the master cylinder pressure acting on the larger end, the control piston moves under the influence of its return spring until both ports of the rocker-type valve are closed. In this condition, the hydraulic pressure in the brake system is the sum of the pressure output from the master cylinder and the additional pressure due to the action of the servo.

If the pedal is then released, the pressure on the larger diameter end of the control piston falls to zero. This piston is then moved by the hydraulic pressure in the brake circuit and its return spring until the rocker valve opens the vacuum port, the air inlet remaining closed. Manifold depression is then communicated to the valve chamber and through the pipe connecting it to the outer end of the servo piston and cylinder assembly. Thus both sides of the servo piston are again subject to manifold depression and the piston is returned by the spring to its outermost position.

On the other hand, if the brake pedal, instead of being released, is depressed further, the hydraulic pressure on the larger diameter end of the control piston is increased. This moves the piston in the opposite direction, once again opening the air inlet port and closing the vacuum port, to increase the pressure on the outer end of the servo piston until the forces on the control piston are balanced and the rocker valve is once more centralized.

Layout of the servo unit

This unit, Figs. 4 and 5, comprises a body casting, on one end of which is secured a pressed steel cup that forms the servo cylinder. The end of this cylinder is closed by an inserted, dished pressing. A compression spring is interposed between the piston, which is also a pressed steel assembly, and the end of the cylinder adjacent to the body casting. The piston rod projects through a guide-bush and a hydraulic seal, housed in the casting, into the chamber to which the connection from the master cylinder is made—this connection is in fact made through another chamber, at the large diameter end of the cylinder for the control piston. When the brakes are off, the conical end of the servo piston rod is just clear of a seating in the output piston, the movement of which is limited by a nylon distance sleeve, or stop, shown on a large scale in the left hand corner of Fig. 5. A barrel-shaped coil spring bears against a seal assembly on the other side of the output piston, tending to retain it against this nylon stop.

The control cylinder, which is shouldered to receive the two different diameter portions of the control piston, is parallel to the cylinder that contains the output piston. Its larger diameter end is closed by a plug and a rubber seal, inserted into the end of the casting and retained by the servo cylinder. The plug also serves as a stop to limit the movement of the control piston in that direction. Movement in the other direction is limited by the abutment of the small diameter end of the control piston against a Belleville washer.

To facilitate production, the control piston is made in two parts, one being the large and the other the small diameter portion. This enables the two components to be centreless-ground. It also obviates the necessity for accurate alignment between the bores of the two portions of the cylinder. The end of the smaller diameter portion is turned down to form the stop that limits its travel in the direction away from the servo cylinder.

The axis of the T-shaped rocker and valve assembly is not horizontal, but slopes downwards from the control piston. This arrangement has been adopted partly for the sake of compactness and partly so that, should any foreign matter enter through the port from atmosphere, it will be trapped in the valve chamber and therefore cannot get into the clearance between the control piston and its cylinder. In addition, this arrangement makes in unlikely that foreign matter will enter the servo cylinder. The valve assembly comprises simply a T-shaped rocker, on the ends of which are secured the nylon valves. This assembly is held down by a hairpin type spring, and located by a top hat section retainer, secured by a screw and Mills pin to the body casting. The ends of the spring serve as the pivot pins, by means of which the valves are mounted on the rocker, and they also prevent the assembly from sliding out of the ends of the top hat section retainer.

DETAILS: Servo piston and cylinder assembly

A simple, 16 S.W.G. mild steel, cup-shaped pressing forms the servo cylinder. Its bore over the length of the working portion is 6·893-6·887 in diameter. The base of the cylinder is dished inwards to increase its rigidity, to locate one end of the piston return spring, and also to make the unit more compact by enabling it to be fitted snugly over the end of the body casting.

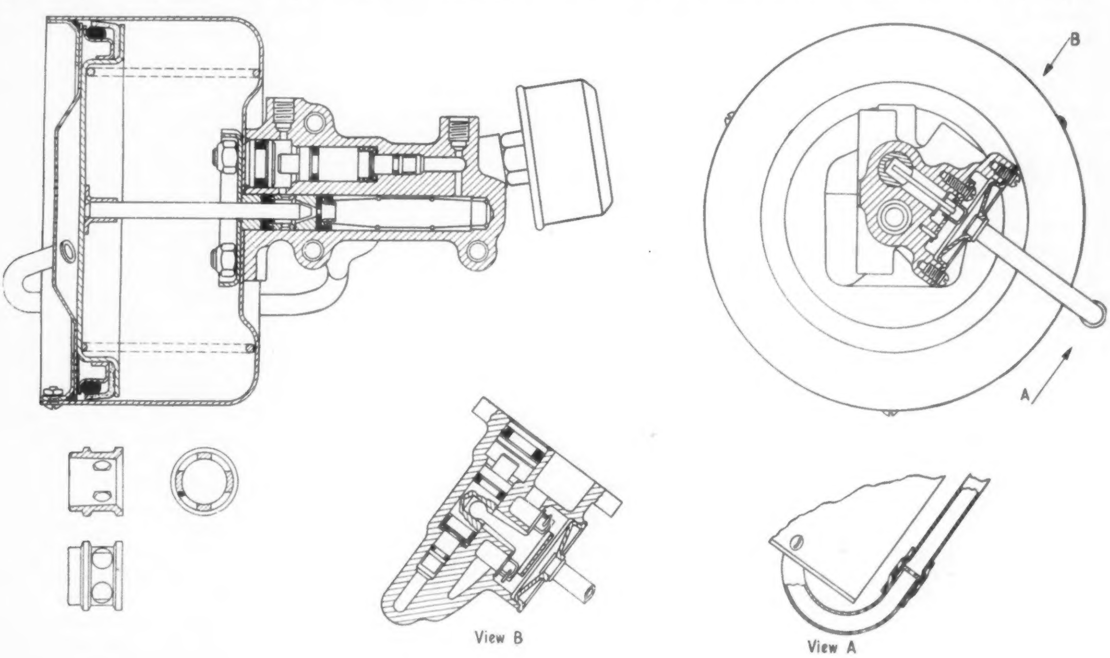
Four $\frac{5}{16}$ in diameter studs on the body casting project through holes in the base of the cylinder and in a 14 S.W.G. dished retainer plate, and the assembly is secured by Pinnacle self-locking nuts. Since a vacuum connection passes through this end of the cylinder, and therefore the joint must be effectively sealed, a $\frac{1}{32}$ in thick Oakenstrong washer is interposed between the cylinder and the casting.

The cylinder is given a primary black oxide anti-corrosive treatment, which helps to key on the secondary treatments. All the external surfaces are coated with black stove enamel. Inside, the bore is coated evenly with Molypol—molybdenum disulphide in suspension in a fluid—and allowed to dry.

Closing the open end of the cylinder is an assembly comprising an 18 S.W.G. dished steel end plate, an annular pressed rim of the same material, and a rubber sealing ring of 0·143-0·135 in diameter circular cross section. Spot welds secure the end plate and rim together and the rubber seal is fitted in the angle between the periphery of the end plate and a chamfer round the rim. The central portion of the end plate is dished, so that the Bundy tube, forming the connection to the valve unit, does not have to be bent through too large an angle. A bead is formed near the end of the tube, which is passed through a hole punched in the end plate and then flared. The whole end-plate assembly is inserted in the end of the cylinder and retained by three screws passed radially through the cylinder wall and the rim. A lip formed round the rim prevents its being pushed too far into the cylinder during assembly. All the steel components are cadmium plated.

The piston is a dished, circular plate, the periphery of which is flanged to carry a seal assembly. The dished portion serves to locate one end of the return spring. At its centre is spot-welded a flanged ferrule, into which the piston rod is pressed. This ferrule is bored after it is welded on the plate; this is to ensure that the rod is held with its axis concentric with that of the remainder of the piston assembly.

Fig. 5. General arrangement of the Girling vacuum-servo brake control. The three views in the bottom left-hand corner are of the nylon distance sleeve



A Cretol, 0·100-0·075 in thick soft leather, cup type seal is employed. It is soaked in Molycue 10 and drained. This molybdenum disulphide treatment of the seal and the bore enables the assembly to be operated indefinitely without any other liquid lubricant or grease, and therefore, obviates maintenance operations. The fluid, in which the molybdenum disulphide is suspended, for treatment of the leather, is carefully chosen to be suitable for operation at low temperatures in cold countries.

A pressed retainer ring holds the seal in position, and a sponge rubber backing ring is interposed between the periphery of the retainer and the flange of the leather seal. This backing ring serves to ensure that the flange of the seal is pressed uniformly against the bore of the cylinder throughout the whole life of the unit. Sponge rubber is used in preference to a steel spring ring, because it ensures that the radial pressure is uniform round the periphery and also because it is more resilient and unlikely to bed into the leather and thus become less effective. All the metal components of this assembly are phosphate-coated and dipped in lanoline to prevent corrosion.

An En1A piston rod is employed, but it is chromium plated and polished, to give a surface finish of 6 micro-in or better, over the length that slides in the case hardened, En32B bush in the end of the body casting. This sliding portion is not lubricated. As has already been mentioned, one end of this 0·311-0·3105 in diameter rod is pressed into the ferrule at the centre of the piston; the other end is chamfered, its included angle being 40 deg.

An 8 S.W.G. return spring is fitted. Its free length is approximately $10\frac{3}{4}$ in and its load, when it is compressed to $3\frac{1}{8}$ in, is 17-22 lb. The total number of coils is $4\frac{3}{4}$ and its ends are not ground. This spring, the outside diameter of which is $5\frac{3}{16}$ in, is phosphate-coated and dipped in lanoline to prevent corrosion.

Body casting and pistons

An LM12-WP aluminium gravity die casting is employed for the body of the unit. It is well cored to reduce the volume of metal. Although the principal advantage of this coring is that porosity, due to excessive shrinkage on cooling, is avoided, it also reduces the weight of metal used and therefore the cost. The casting has been designed in such a way as to minimize machining operations.

The four $\frac{5}{16}$ in diameter studs that secure the servo piston assembly are cast in the end face of the body. They have square heads formed on their ends to key them into the aluminium. The heads measure $\frac{3}{8}$ in across the flats and the shanks have a 24 TPL ANFS thread machined on them. The end face is not machined, but as has already been mentioned, the joint between it and the servo cylinder is sealed with an Oakenstrong washer.

All the bores for the pistons are honed and Faessler rolled, to give a surface finish of 12 micro-in or better. This treatment is important because it is essential that the pistons move freely in their bores; moreover, the valve control piston operates virtually without any lubricant. A tolerance of 0·001 in is allowed on the bores and 0·002 in on the pistons, giving a maximum clearance of 0·0035 in and a minimum clearance of 0·0015 in. The only other machining done on the casting is spot facing for the valves, and drilling and tapping for the mounting studs, for the screws that secure the valve chamber cover, and for the hydraulic connections. The casting is Pyluminized before machining, to protect it against corrosion.

The cylinder for the hydraulic output piston is of uniform diameter from its outer end to the innermost extent of its working length. This is a good feature from the point of view of ease of production. The end of this cylinder is closed by a case hardened En32B bush, which, as has already been mentioned, serves as a guide for the piston rod of the vacuum servo. This bush is of plain cylindrical section and is cadmium plated. It projects from the end of this bore to serve as a spigot to locate the vacuum-servo cylinder. A channel section hydraulic seal, made of natural rubber, is assembled against its inner face.

One end of a nylon distance tube seats in the channel section seal. The other end of this tube serves as a return stop for the hydraulic output piston. The reason why nylon is employed for this component is that it has good wear resistance, while at the same time its resilience ensures that there is no noise when the piston is returned rapidly on to its end face. Furthermore, because of the relatively complex shape of this small part, it would have been more expensive to produce if it were made of metal. In nylon it can be injection-moulded and requires no machining or other finishing operation. At the end against which the output piston stops, the tube is flanged and about $\frac{1}{8}$ in from its other end a collar is formed. The flange and collar serve to centralize the tube, so that it seats properly in the channel section seal, and the annular groove between them, together with four radial holes in its base, form the passages through which the hydraulic fluid flows to and from the master cylinder.

The piston is of case hardened En32B steel. It is of plain cylindrical section, but has a conical hole through its centre. This hole is 0·174-0·170 in diameter at its smallest end, and the cone angle is 40 deg to 40 deg 30 min; it forms the seating for the end of the servo piston rod. The outer periphery, which is 0·624-0·622 in diameter, is ground and polished.

A cup-shaped, natural rubber seal seats against the inner face of the piston. This seal has a 0·110 in diameter hole through its centre. The end of the conical face on the servo piston rod is of such a diameter that, when the servo unit is in operation, it projects through the conical hole in the hydraulic output piston and seats on this rubber seal. A short tube, with a flange at one end, is assembled into the seal. The other end of this tube spigots into the return spring, the seating for which is formed by the flange.

The En49C spring is of barrel-shape. This is because it is 3 in long when fitted and therefore requires to be stabilized. The large diameter portion of the spring is centralized in the bore, while its smaller diameter ends are of such a size that they fit readily on to their seatings. In the free condition, the spring is $4\frac{5}{8}$ in long; a load of 3 lb is required to compress it to its fitted length. It is phosphate-treated for protection against corrosion. The employment of this treatment, in preference to cadmium coating, obviates any danger of hydrogen cracking.

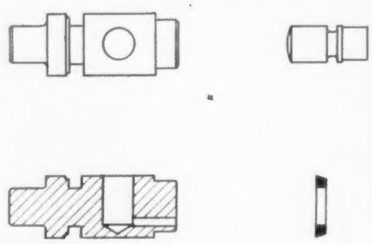


Fig. 6. The two-piece control piston. Below are shown cross sections of the larger portion and its taper-faced rubber seal

Control cylinder and piston

Unlike the cylinder for the output piston, that for the valve control piston is shouldered so that it has three different diameters, one for the small end of the control piston, the second for the large diameter end and the third for the chamber between the large diameter end and the servo cylinder. The end of this chamber is closed by a mild

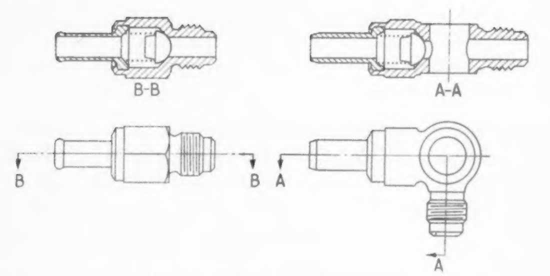


Fig. 7. Plan and section of each of the alternative non-return valves

steel plug of plain cylindrical section with an annular groove round it to carry the natural rubber seal. This seal is of approximately rectangular section, but has a tapered peripheral face. The outer end of the plug bears against the base of the servo cylinder and its inner end forms a stop for the valve control piston. To ensure that this plug is adequately supported, the flange of the stiffening plate inside the servo cylinder is so arranged that it passes over the end of the plug.

A case hardened En32B valve control piston, Fig. 6, is employed. As has already been stated, it is in two pieces; this is to obviate the need for accurate alignment of the bores in which the larger and smaller diameter portions operate, and also to facilitate production of the component. The ratios of the diameters of the two portions is 2·8:1 and this, together with the rate of the light return spring, determines the ratio of servo assistance: physical effort. One end of the piston is turned down to $\frac{3}{8}$ in diameter to form the abutment against the plug when the valve is at one extreme of its travel; the other end is reduced to 0·480-0·490 in, where it spigots into the return spring. The larger portion is $\frac{5}{8}$ in diameter. Both portions of the piston are chemically treated to give a black oxide surface finish for protection against corrosion.

Round the periphery, near the end adjacent to the stop, there is an annular groove to carry a taper-faced rubber seal. With this type of seal, hydraulic pressure tends to force the acute-angle end of the periphery out against the bore. Thus, the degree of sealing, or radial, pressure increases in direct proportion to the hydraulic pressure that has to be contained. This shape is obviously such as to give a better seal than, for example, an O-ring, since a hydraulic wedge might be formed at the leading edge of an O-ring as it moves along the bore of the cylinder, and this might cause slight leakage. Another advantage of the taper-faced seal is that the area of contact between the seal and the bore is small and, therefore, friction is minimized.

An interesting detail feature of the arrangement is that the corner between the outer periphery of the side wall of the groove and the land, adjacent to the acute angle of the taper-faced seal, is relieved. This is done by chamfering, or undercutting, the corner with a $\frac{1}{32}$ in radius cutting tool, so that it is concave instead of convex. The undercut, of course, is incorporated to accommodate the edge of the seal when it is deflected on being pressed into the bore. If this measure were not taken, there would be a danger that the edge of the seal might tend to jam in the clearance between the piston and its bore.

Between the seal and the abutment for the return spring, a 0·3135-0·3115 in diameter hole is drilled and reamed radially into the piston. This hole receives the end of the T-shaped valve rocker. A $\frac{3}{32}$ in diameter hole is drilled axially from the adjacent end of the piston to break into the radial hole. This, of course, is to prevent pressure changes, as the piston moves axially, in the large diameter portion of the bore that houses the return spring.

The return spring, which is spigoted on to the end of the large diameter portion of the control piston, is designed to offer a compressive resistance of 0·6-0·5 lb when its length is $\frac{7}{8}$ in, and 3·6-3·5 lb when its length is $\frac{3}{8}$ in. It is phosphate-treated for protection against corrosion. This spring seats on a nylon washer in the form of an auxiliary spring of the Belleville type. The function of the nylon auxiliary spring is to prevent a very abrupt change in the output characteristic when the coil spring goes solid. It seats on the shoulder between the bores for the large and small diameter portions of the control piston.

On the small diameter portion of the control piston, the end adjacent to the large diameter portion is domed. This arrangement has been adopted to ensure that the load transmitted between the large and small diameter portions is truly axial. The working portion of the smaller component is 0·374-0·373 in diameter, and the end remote from the large diameter portion is turned down to 0·350-0·340 in diameter to clear the unfinished length at the end of the bore and to act as a stop to limit the motion in that direction. Yet another extension, 0·115-0·125 in diameter, is incorporated so that it is impossible to assemble the component incorrectly. A groove for an O-ring is machined round the end of the working periphery, adjacent to the stop. The reason why an O-ring is used on this component is that it would be difficult to produce a taper-faced seal of such small dimensions. Since the volume of rubber in the O-ring is so small, it is necessary, of course, to manufacture the component to close tolerances to ensure that it is not over-stressed when it is compressed in its groove. The friction at the working faces is not great because the ring is so small.

Valve assembly

The valve rocker is a simple 14 S.W.G., cadmium plated, T-piece. Its leg is tapered and the extreme end is circular in shape, as viewed in elevation. Each side of the circular portion is coined to a 0·311-0·305 in spherical diameter, so that it will fit snugly in the radial hole in the control piston. A 0·052 in diameter hole is pierced through each arm of the T-piece to receive the ends of the hairpin type, valve-return spring, which also retains the two nylon valves. Not only are nylon valves less expensive than components machined from metal and faced with rubber, but also this material is noted for its good wear resistance, strength and resilience. Plate type valves, $\frac{1}{2}$ in diameter, are employed. Formed on one side of each are two lugs, between which the arm or the T-shaped rocker fits. The ends of the return spring are passed through holes in the lugs and the arms of the T-piece.

An 18 S.W.G., cadmium plated, top hat section pressing retains the whole assembly. Its length is only slightly less than the distance between the holes in the arms of the T-piece and the valve lugs, so its ends fit over the lugs and serve to locate the valve assembly between the ends of the hairpin spring. The centre of the hairpin spring is bent to form an eye, which is clamped between the flange of the top hat section and the body casting. A 2BA round-headed set screw is passed through this flange and screwed into the casting to hold the assembly down, and a $\frac{1}{8}$ in diameter Mills pin is driven through a hole in the opposite flange into the casting. The spring is of 18 S.W.G., En49 Grade II.

The valve chamber is closed by a spigoted-in, moulded nylon cap, which is secured by two small set screws passed through its flanges and into the casting. A 1 deg taper is incorporated on the spigot portion of the cover the largest diameter being adjacent to the flange. Nylon is employed for this component because of its resilience, strength and resistance to deterioration. Resilience is desirable, so that the alignment of the Bundy tube connecting the chamber to the opposite side of the servo piston is not critical. This tube is beaded near its end, inserted through a hole in the

centre of the nylon cover and flared, the grip between the flared end and the beading being adequate to form an air-tight seal.

A number of precautions have to be observed in the design and manufacture of nylon components. For example, it is necessary to allow for dimensional changes after moulding. The moulding process tends to drive off most of the moisture content of the nylon and subsequently, as moisture is reabsorbed, dimensional changes occur. At 70 per cent relative humidity, the equilibrium value of the water content is approximately 4 per cent. In general, during a change in water content from 2 per cent to 4 per cent, a component of this form can be expected to increase in size 0·003 in per inch diameter per 1 per cent change in water content.

An annealing treatment is generally desirable to relieve the strains introduced by moulding. This process may consist of heating the mouldings in a high flash point, non-oxydizing oil, to a temperature at least 20 deg C higher than its expected maximum temperature in service. Where the maximum service temperature is not known, the mouldings can be annealed at a temperature of 170 deg C. The re-humidification process is carried out after annealing; it consists in immersing the components in hot water until a moisture content of 4 per cent is attained. The period of immersion is initially determined experimentally.

A restriction is incorporated in the pipeline between the valve chamber and the vacuum-servo cylinder. Its function is to restrict the flow of air from the servo cylinder and thus to damp the motion of the servo piston when the brakes are applied: this ensures that the piston rod comes up gently against the hydraulic piston without making any noise. The restrictor is simply a plain washer with a $\frac{1}{8}$ in diameter hole in its centre. It is assembled into the junction between the Bundy tubes connected to the end plate of the cylinder and the valve chamber. The diameter of one of these two tubes is increased, so that it forms a socket to receive the restriction washer and the end of the other tube. The joint is sealed with a rubber sleeve, fitted externally. Synthetic rubber is employed at this point because of its resistance to deterioration due to contamination by fuels or lubricants, which might inadvertently get on to it.

Air cleaner and other accessories

A small air cleaner is fitted to the end of the air intake passage. Externally it is stove enamelled. A special enamel resistant to deterioration, due to contact with brake fluid, is used both on this air cleaner and for the servo cylinder. The filter is similar to those employed for engine breathers. It is packed with 33 S.W.G. galvanized steel wire, knitted and crimped. Before assembly, it is immersed in brake fluid and the surplus drained off. The manufacturers recommend that every 5,000 miles it be washed in methylated spirits and then soaked again in brake fluid.

It is advisable, though perhaps not essential, that a vacuum reservoir be incorporated between the servo unit and the manifold. Whether or not a reservoir is fitted, a spring-loaded, non-return valve, Fig. 7, should be incorporated in the pipeline. This non-return valve serves two purposes. It prevents damage to the servo unit in the event of a blow-back in the engine induction manifold. Secondly, it prevents the unit from being contaminated with petrol if the induction manifold should be flooded during cold starting. Two non-return valve assemblies are made by Girling; one is for use in systems that do not incorporate a reservoir, while the other is a banjo unit for attachment to a reservoir.

This servo unit, both in broad principle and in detail, is a first-class example of sound engineering design. A note-worthy feature is that costs have been reduced to a minimum without sacrifice in quality. The unit has certainly been designed to last the life of the vehicle. Operational experience has shown that the unit will function effectively for very long periods without any attention at all; despite this, the manufacturers do recommend that the air filter be cleaned regularly.

Ride Control

A NEW device, the "Load-Leveller," to prevent sagging rear springs in heavily loaded passenger cars, to neutralize bucking on sudden stops and to prevent sway when cornering, has been developed by the Monroe Auto Equipment Co., of Monroe, Michigan, U.S.A. It is claimed that the device will increase the road clearance of a heavily loaded automobile from 35 to 40 per cent, and the clearance of a car with a normal load is increased from 12 to 17 per cent.

The "Load-Leveller" combines an oversize, calibrated shock absorber with an overload spring. It replaces the rear shock absorbers. Prolonged testing has amply demonstrated that a four-door sedan equipped with this new product can be loaded with six passengers and 500 lb of luggage and still maintain a level and comfortable ride. On the same car with only one or two passengers and no luggage, the Monroe "Load-Leveller" ensures a non-sway, stabilized ride.

Several advantages are claimed for this new product, which, by means of a calibrated control system, gives automatic compensation for overloading. For example, stability on curves is increased and a correct distribution of weight is maintained to prevent changes in wheel castor that might make steering more difficult and cause excessive tyre wear. It also ensures that the headlight beam is maintained at the correct angle when the car is heavily laden. This, of course, will make for much safer night driving.

Nitrile Gum Plastics

THE gum plastics composition Royalite is being increasingly used by the automobile industry in the United States of America. It was developed by the United States Rubber Company and is now being manufactured in this country by The North British Rubber Co. Ltd., 62/64 Horseferry Road, London, S.W.1. There are two basic types of Royalite, rigid and flexible. They are formulated to manufacture products that will give adequate service in the temperature range −20 deg F to approximately 240 deg F.

This material is now being used by General Motors, Ford, Chrysler and Studebaker-Packard for a variety of inside body parts. It is, for example, used on several current models for seat side shields. Through its use, shapes that were previously impossible are now being produced with ease. By a simple vacuum-forming operation it is possible to produce a part, which previously had been fabricated in steel and hand-trimmed in leather or fabric, with definite limitations on design.

Royalite is also being used for instrument panel trim. On a De Soto model a flexible Royalite cover was applied directly over the metal. An interesting feature of this application is that the actual metal stamping was used as part of the vacuum-forming mould, with the Royalite being cemented directly to the metal by means of the heat-activated cement. Other applications include door trim pads and centre pillar covers for four-door hardtop models.

A hydraulic torque converter for the Borg-Warner automatic transmission

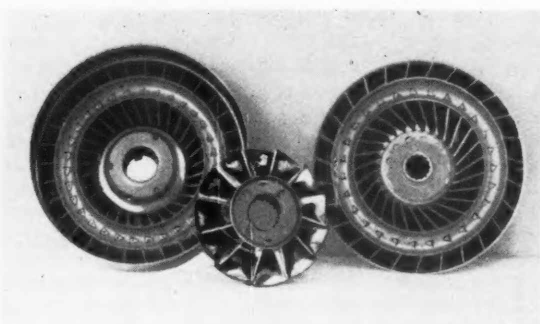
HYDRAULIC TORQUE CONVERTERS

The Manufacture of Units for Borg-Warner Automatic Transmissions

WHEN about a year ago, Borg-Warner Ltd., Letchworth, started the manufacture of automatic transmissions, the factory was equipped for the production of only the gearbox. The hydraulic torque converter which is, of course, an essential part of the Borg-Warner system had to come from America, until such times as the necessary equipment could be obtained and installed. Now the torque converter section at Letchworth is in operation and the Borg-Warner automatic transmission used on British and Continental cars is produced completely in this country.

Essentially, the Borg-Warner hydraulic torque converter may be said to comprise an impeller assembly, a turbine assembly, a stator assembly, a direct drive clutch assembly, a flywheel assembly and a cooling fan or blower. The problems involved in producing these various parts differ from those generally met in automobile manufacture. The chief difficulty arises from the fact that a certain amount of welding must be carried out on parts that because of their size and form are strongly susceptible to distortion, and for two reasons distortion must be avoided. First, on many elements close dimensional tolerances must be maintained; second, because of the high rotary speeds at which the hydraulic converters work, only a very small departure from true balance can be tolerated. All the assemblies are in greater or lesser degree based upon pressings, which are obtained from outside suppliers.

The impeller, stator and turbine assemblies

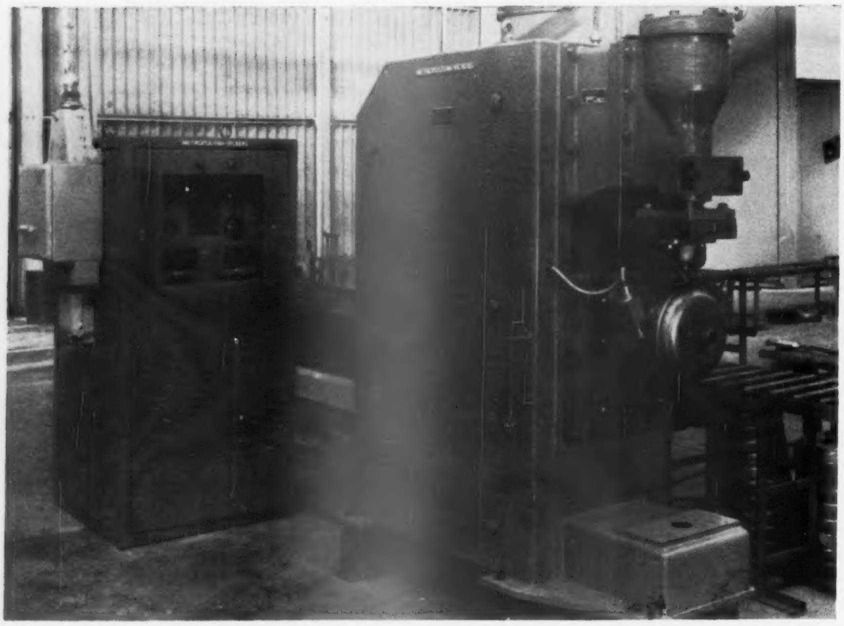


In the layout of the hydraulic torque converter section, and, indeed, in the layout of the whole automatic transmission factory, Borg-Warner Ltd., have acted on the assumption that the demand for fully automatic transmissions will increase as their value becomes more generally recognized. The layout is, therefore, designed for continuous flow production, but until the demand increases to something approaching the production capacity, batch production is being employed. In short, the aim is at maximum labour utilization rather than maximum machine utilization. This is effected by having one operator process a batch of parts through several successive operations.

The impeller

The impeller is built up from a pressed steel shell of torus form, a hub of hardening quality carbon steel, 31 pressed steel vanes, a steel hub plate and a core ring. At the first operation the hub is pressed into the shell on an arbor press. This assembly is then transferred to a Metrovick 600 kVA projection welder where the hub plate is fitted over the hub on the inside of the shell for projection welding to the shell. After this welding, the hub is re-pressed in the shell and is then welded to the outside of the shell by submerged arc welding. Initially it was found difficult to prevent hardening of the hub during the weld cycle, since the material is En 47 or other medium carbon steel. This difficulty was later overcome by fitting a split brass bush into the hub bore for the duration of the cycle.

At this stage the impeller is ready for the insertion of the 31 vanes, an operation that is carried out on a small hand



This Metrovick machine is used for welding the drain and filler plug adaptor and, diametrically opposite, the counterweight to the impeller shell

press. The shell is mounted in a contoured fixture and the vanes are inserted singly. This is a type of operation it would be difficult to mechanize. As received, the vane is so formed that it must be sprung into the slots in the impeller. It will be seen that a complex and sensitive mechanism would be needed if the operation were to be automatic instead of manual.

The pressed steel core ring is next inserted. A small tab on each vane projects through the core ring and is folded over on a 12-ton press. Two strokes of the press complete this operation. Further welding is next carried out on a machine with an indexing fixture. First, an adaptor to take the filler and drain plug is welded in to a hole in the shell. The work is then indexed through 180 deg and a counterbalance weight is welded on the outside of the shell diametrically opposite the adaptor. The welding current necessary for the adaptor differs from that needed for the counterbalance weight, and indexing the work from one position to the other automatically effects the necessary switching on the welding machine.

Projection welding the hub plate to the impeller hub

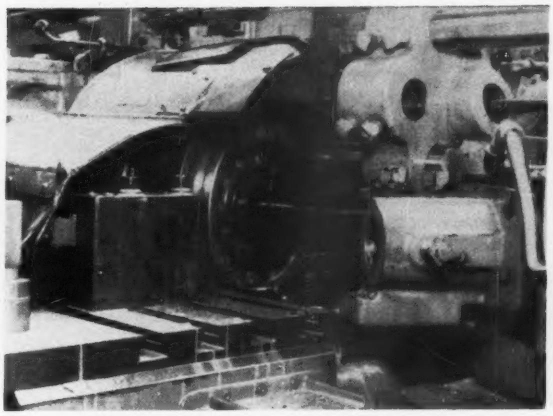
Security of the core ring is important, and at the next operation a contoured tool in a 75-ton press stakes the vane tabs in the core ring. All 31 vanes are staked at a single stroke. For further security, the tabs are then rolled in the ring. For this operation the work is mounted on the bed of a drill press. A special tool, carrying two rollers, spaced at the pitch circle of the core ring, is mounted on the drill spindle. The spindle is brought down and the rollers are held under pressure against the vane tabs for three or four revolutions.

Two fairly complex machining operations are then carried out successively on Gisholt Fastermatic lathes. On the first of these machines, the impeller is mounted in an air chuck with its open end offered to the tools. The special chuck jaws grip on the outside diameter of the shell. At this setting the hub bore is rough and finish machined, the hub plate and the open end of the shell are faced and the corners broken, and a short length of the outside diameter from the open end of the shell is rough turned and chamfered.

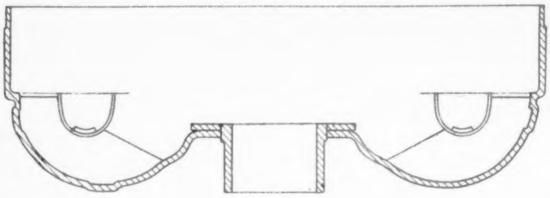
On the second Fastermatic machine the impeller is mounted with the hub end offered to the tools. Location is taken from the hub bore and the machined open end of the shell. An air-operated drawbar pulls the work against the machine face plate. The work is held in a special chuck, while 12 spring-loaded buttons contact the inside diameter of the shell directly opposite the tools. These buttons eliminate any danger of distortion through either chucking pressure or cutting pressure. At this setting the portion of the outside diameter rough turned at the previous operation is finish turned, two diameters are turned on the hub and the hub end is chamfered.

When the impeller assembly leaves the second Fastermatic, the drain plug and its seal are fitted, preparatory for the first leakage test. Freedom from leakage is absolutely essential. The impeller is mounted in a fixture that seals the open end of the shell and a rubber plug is inserted to seal the hub bore. The fixture and the work are submerged in water, and air at 80 lb/in^2 is pumped into the shell to check for leaks, particularly at the hub weld and at the adaptor and drain plug.

The impeller vanes are radial, and, therefore, provide little reinforcing effect to prevent distortion of the shell at high speeds. To avoid any risk that loosening will occur due to such distortion, 11 of the 31 vanes are brazed into place. Brazing is done by hand with an oxy-acetylene torch. The work is mounted in a fixture that is arranged for manual indexing. Following the brazing, the work is pickled, washed and dipped in an anti-rust solution.



The set-up on a Gisholt Fastermatic lathe for the first machining operation on the impeller shell. The work is held in an air chuck with specially-designed jaws



The impeller shell is of torus shape. This sketch also shows the vanes and the core ring

The outside diameter of the impeller hub forms a journal to run in a supporting bearing. To give this journal good wearing qualities it is necessary to harden the hub for a length of 0·750 in to a minimum depth of 0·015 in. This hardening is carried out automatically on a Wild-Barfield induction hardening machine.

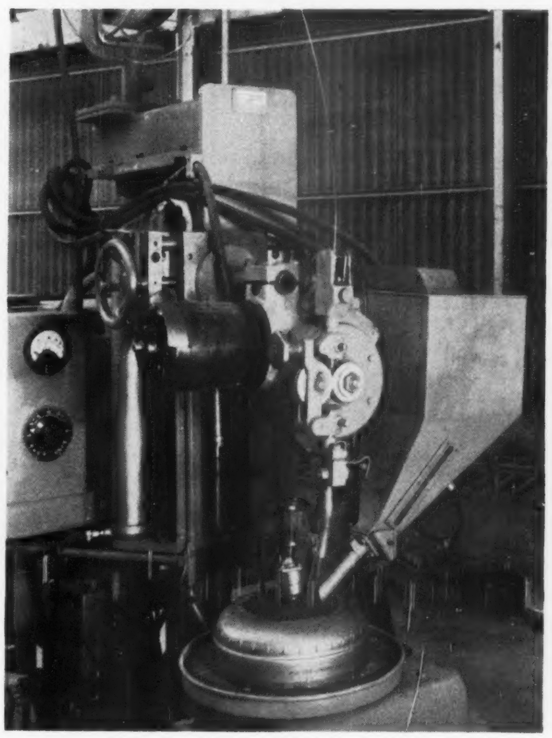
The impeller hub drives the front transmission pump. To provide this drive, the end of the hub is straddle milled to produce two diametrically opposed tangs. To complete the impeller assembly, the outside diameter of the hub, where it engages the main oil seal, is ground to a finish of 15 micro-inches. The work is mounted on a face plate fixture in a Schaudt U.R. grinding machine with location taken from the hub bore and the hub face plate. It is not possible to employ normal cylindrical grinding practice because of the torus shape of the shell. The grinding wheel is mounted at an angle in relation to the axis of the impeller and the wheel face is dressed to be parallel with the work axis. To all intents and purposes plunge grinding is carried out. This completes the production operations on the impeller assembly, but every impeller is given a complete check before it is transferred to the final assembly section. All machined dimensions are checked, and in addition there are checks for run-out on the turned outside diameter of the impeller shell, the outside diameter of the hub and the face of the hub plate. After inspection the assembly is washed ready for final assembly.

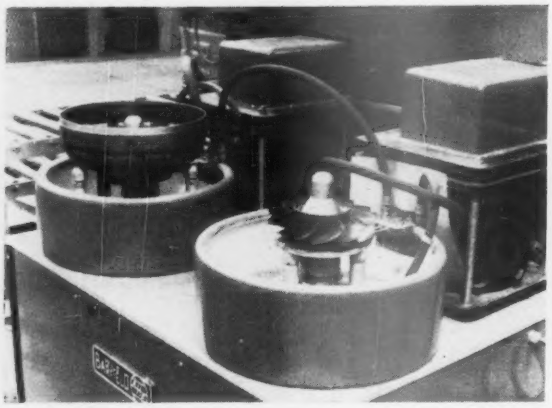
The turbine

In that it comprises a pressed steel shell, vanes, a core ring and a hub, the turbine assembly resembles the impeller. It does not, however, call for so lengthy an operation sequence. In fact, there are neither machining nor welding operations in the production sequence. The vanes, in this case 27 in number, are curved and their leading and trailing edges are coined sharp. They are inserted in the same manner as the impeller vanes, with the difference that the slots in the turbine shell pass right through and the vane tabs are folded against the outside of the shell. The vanes are folded on a 30-ton crank press and the folding of the vane tabs in the core ring is effected in a 12-ton hydraulic press. The turbine hub, which is received in a fully machined condition, is riveted to the shell. There are 12 rivets, and riveting is carried out in a crank press equipped with a shuttle punch and die.

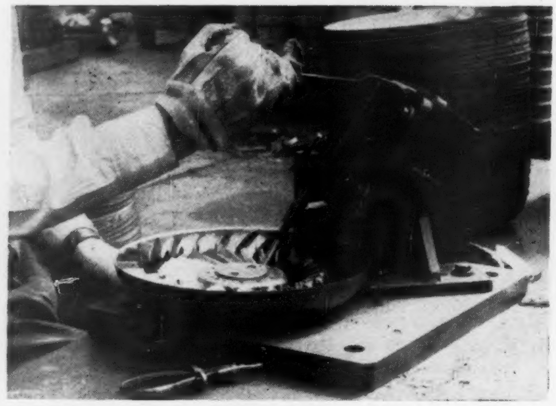
As there is no machining on this assembly, it is necessary to check for run-out after the riveting operation. It is also necessary to balance the turbine to within 0·1 oz-in. Out-of-balance is corrected by welding on a strip of metal of appropriate weight. A welding gun is mounted immediately adjacent to the Jackson and Bradwell balancing machine.

Submerged arc welding is used for joining the hub to the impeller shell





A two-station Wild-Barfield machine for induction hardening the outside diameter of the impeller hub, left, and the bore of the stator hub, right



Blades are inserted in the turbine shell on this hand press. A similar type of press is used for inserting blades in the impeller shell

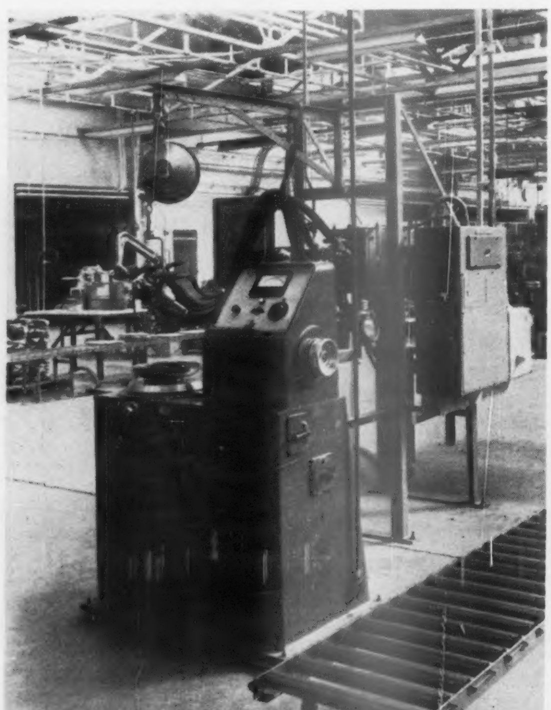
Stator

Stators are received from outside suppliers with the vanes inserted and the hub part-machined. The first operation on the stator at the Borg-Warner factory is to copper braze the vanes to the hub. Brazing is carried out in a Wild-Barfield continuous furnace with a protective atmosphere. After brazing, the stator is transferred to an Ex-Cell-O fine borer tooled to machine the bore to a total tolerance of 0·002 in, to face both sides and to cut off the tabs from the 12 vanes.

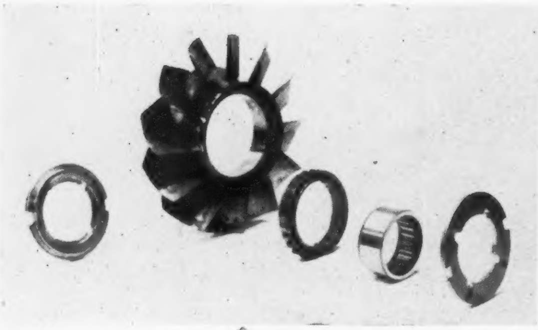
The bore of the stator hub acts as the outer race for the stator freewheel, or one-way clutch. It is therefore, necessary to introduce a hardening operation, which is carried out on the second station of the Wild-Barfield induction hardening machine that is also used for hardening the impeller hub. In this case, a minimum length of 0·500 in is hardened to a depth of 0·060/0·100 in to Rockwell C60/64. Within 15 minutes of being induction hardened the stator must be stress relieved. For this, it is passed through a Holcroft draw furnace for two hours at 275 degF.

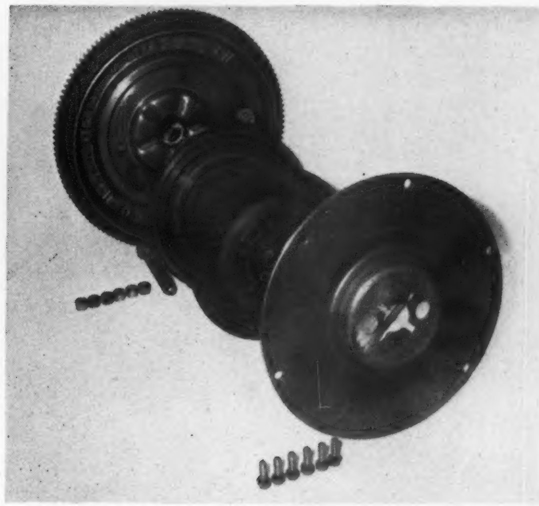
Immediately after being stress relieved the stator is washed and rust-proofed. Two grinding operations follow. At the first one face is ground. For the second grinding operation the stator is reversed in the machine chuck and the other face is ground and the internal diameter is finish ground to a diametral tolerance of 0·0005 in and to a finish not exceeding 10 micro-inches. All this grinding is carried out on one machine, a Schaudt internal grinder. The procedure is to grind a batch on one face and then follow on by grinding the other face and the internal diameter.

Machine for checking the balance of the turbine assembly. The welding equipment for correcting out-of-balance is adjacent



Out-of-balance in the turbine assembly is corrected by spot welding the appropriate weight on to a vane



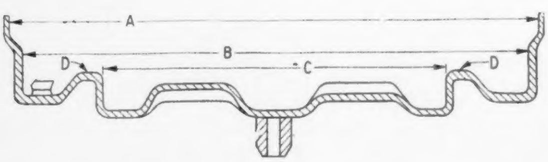


Above: Components for the stator assembly
Right: Components for the direct drive clutch assembly

Following the grinding operations the stator is washed and is then tested for balance. As with the turbine, it must be in balance within 0·1 oz-in. Correction for out-of-balance is effected by welding an appropriate weight to a vane. The welding is carried out on a 50 kVA machine adjacent to the balancing machine. Every stator is given a thorough inspection before it is released to final assembly. Included in the inspection sequence is a special indicating fixture for checking hub face run-out, hub width and vane form clearance.

Flywheel

The flywheel is an intricate pressing. It is received with the piston fixing brackets already welded in position, but before it is ready for assembly it requires a fair amount of machining to close limits. The first operation is carried out on a Gisholt No. 12 auto lathe. On this machine the pressing is chucked on the outside diameter of the pilot against a magnetic face plate fixture. At this setting three internal diameters, shown at A, B, and C, in the accompanying sketch, are machined. The diameters A and B are finish machined, while C, which, for functional reasons must have a very high standard of surface finish, is rough machined.



This sketch shows the form of the flywheel shell

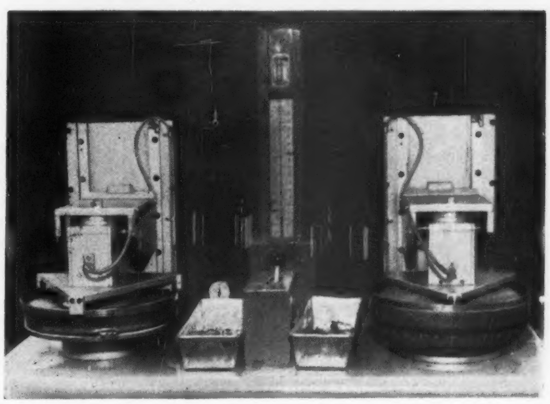
At the same time an undercut is machined in A, the piston fixing brackets are faced, the end of the shell is faced and chamfered, and the face D is faced and chamfered. This operation is carried out on a batch of flywheels, and the machine is then reset for finish boring the diameter C to a tolerance of 0·001 in.

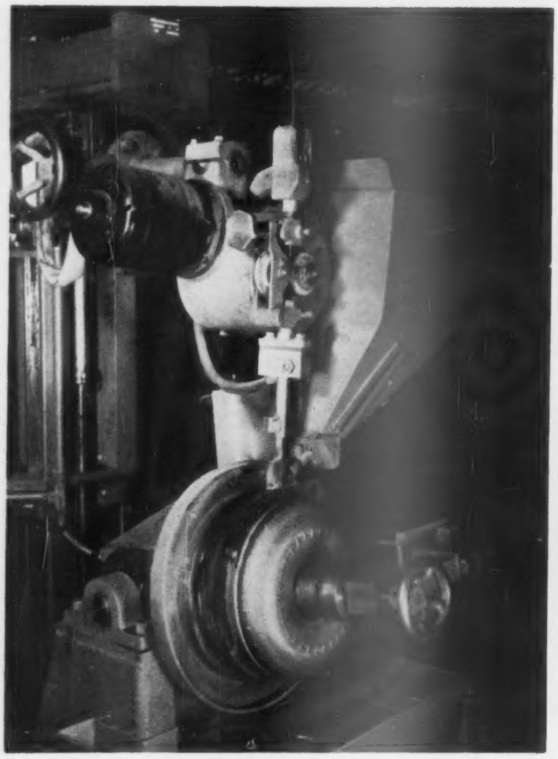
The next operation, referred to as "bearingizing", is for imparting a very high standard of surface finish to the diameter C. For doing this, a tool made of a number of rollers is mounted on the spindle of a drill press and at 200 r.p.m. is quickly fed in to and out of the bore. This slightly increases the bore diameter and gives a highly polished finish. To complete the machining operations the outside diameter of the pilot is ground to a tolerance of 0·0005 in and 15 micro-inches surface finish. As with the other components, every flywheel is completely inspected before it is released for final assembly.

Direct drive clutch

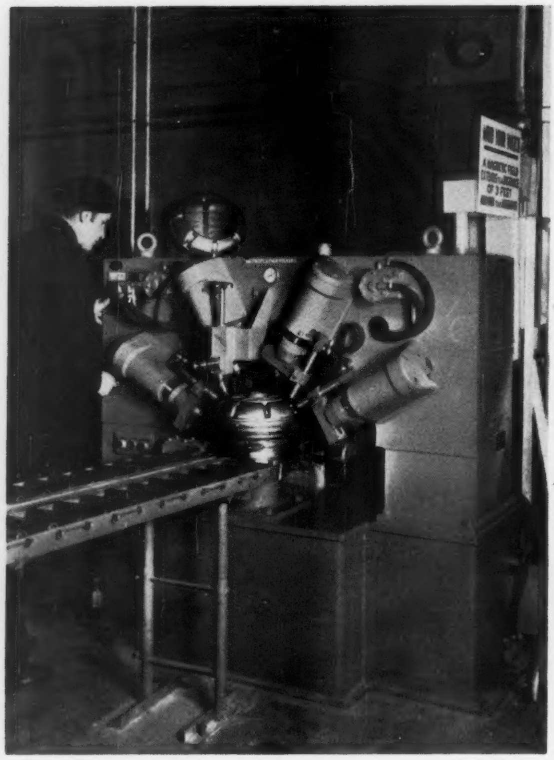
The direct drive clutch unit comprises a piston and a single-disc clutch that is very similar to the normal dry plate clutch. Interest in the production of this unit lies in the checks that are carried out to ensure the requisite high standard of performance rather than in the various production operations. For example, the piston is checked for balance within 0·25 oz-in. So far as the driven member of the unit is concerned, three tests may be mentioned. At the first of these, referred to as a stabilizing operation, the friction facing is clamped and held fast while the hub is mounted on the splines of an oscillating spindle. For 15 minutes the hub is oscillated at the rate of 1,200 oscillations per minute. Following this there is a test for friction lag. In this test the work is mounted on a splined arbor and the friction plate is clamped. A 50 lb load is applied and the beam of the test machine is rotated until the drive sides of the slots contact the stop pins. The beam is then released, rotated in the opposite direction and then allowed to find the mean rest position. At this point the indicator on the test rig is set to the zero position and additional torque is applied to produce 0·150 in indicator movement. All load, except the initially applied 50 lb, is then slowly removed, and the indicator reading obtained gives the friction lag. After the

Special Sigma equipment directly determines the shim thickness to give the correct clearance at final assembly





Submerged arc welding is used for joining the impeller, stator and turbine assembly to the flywheel and clutch assembly



This Metrovick multi-spot welder is used to attach the blower or cooling fan to the complete assembly

tests, this component is also balanced to 0·1 oz-in.

Here it may be remarked that modification of the direct drive clutch assembly is one way in which the transmission is varied to suit the different characteristics of the wide variety of vehicles to which the Borg-Warner automatic transmission is fitted. Substantially the transmission mechanism is the same for all vehicles, and variations to suit the differing characteristics are effected by, among other things, modifications to the friction facing of the direct drive clutch. For some applications the contacting face is left plain; for others it is found advisable to have a number of radial slots in the face; for others, a number of concentric grooves in the face give optimum results; for yet others, a combination of radial slots and concentric grooves gives best results.

Assembly

In the matter of assembly two preliminary points may be mentioned. The first is the scrupulous care to see that only thoroughly clean parts are delivered to the assembly section; the second, the thorough inspection carried out before assembly. This inspection is important since should any fault be disclosed after final assembly, it is necessary to break a weld before rectification can be carried out.

Briefly, assembly follows the following sequence. First the freewheel, or one-way clutch, is assembled in the bore of the stator. Every stator and freewheel sub-assembly is tested to ensure that rotation is in the correct direction and that the freewheel is free running. This sub-assembly is assembled with the impeller, and both are then assembled with the turbine.

In the flywheel and clutch assembly, the first stage is to fit the piston and drive straps to the flywheel. In making this assembly a lip seal and an O ring are fitted. Every part assembly is tested under air pressure for leaks at the lip seal and O ring. The remainder of the direct drive clutch is then assembled into the flywheel.

At this stage the parts are ready for final assembly. Before they are assembled together, a flywheel and clutch assembly and a converter assembly are mounted on a special Sigma comparator. This instrument has two measuring heads and gives direct reading for the thickness of shim required to give the necessary clearance. The two assemblies are then fitted and welded together on a submerged arc welder. Every assembly is tested for leaks. The blower is then welded on, 16 spot welds being made on a multi-spot welder.

Two further operations complete the production sequence. At the first, the assembly is mounted on a Ryder vertical automatic for turning, facing and chamtering the mounting flange; at the second, the complete assembly is balanced. Reference has already been made to balancing of sub-assemblies to tolerances as low as 0·1 oz-in. At the final balancing the tolerance allowed is 0·6 oz-in.

At the final inspection stage, every assembly is tested for free running and for run-out on the face and outside diameter of the mounting flange. Finally, a stall test is carried out on a large percentage of converters. This test consists of driving the converter input member with the output member held stationary. The input torque is adjusted to a standard datum figure, and the speed at which this datum is attained is called the stall speed. The relation between input torque and speed, with the output stalled, is an excellent guide to the correct functioning of the converter assembly. Torque and stall speed vary, of course, for each different engine, but for convenience the production testing at Borg-Warner is done, as mentioned, at a constant input torque; actually about 200 lb-ft.

In addition to all the above testing, each day a number of complete automatic transmissions are given a comprehensive road test on the company's private test track.

THE OLDHAM "Pg" BATTERY

New Constructional Features in a Unit Specially Suitable for Traction and Industrial Purposes

DEVELOPMENT of the electric storage battery has tended to follow three main lines; to increase capacity for a given weight and a given volume and to establish an appreciably longer working life. The capacity of a battery for any given physical size is, in the main, determined by the amount of active material which can be contained within the plate groups. The life of a battery is dependent upon the resistance of the component parts to oxidation and deterioration in acid, and also upon the ability of the grid structure to retain the active material, particularly in the case of the positive plates.

To render the component parts more resistant to the effects of continuous immersion in the acid solution and to the wide temperature variations experienced, numerous improvements have already been introduced into the Oldham battery. The "X" metal alloy of which the grids are made, and the Fibrak separators may be mentioned as two of the developments which have contributed to increased life in service. To ensure the more secure retention of the active material, vitreous felt retainer mats, positioned on each side of a normal flat positive plate, have proved effective, as the vitreous felt becomes embedded in the active material. This reinforcement of the active material offers greater resistance to shedding due to vibration in service.

The slit-type tubular positive plate, whether of ebonite or other plastic material, has also been successful, but to obtain reasonable mechanical strength the wall thickness of the tube is such that the space available for active material is somewhat restricted. Furthermore, ebonite tends to become brittle from the effects of long immersion in acid at varying temperatures. In order to obtain maximum performance it is necessary that as much of the active material as possible is exposed to the electrolyte, but with the conventional ebonite or plastic tube the need to preserve reasonable mechanical strength is a limiting factor in the amount of surface area that may be removed. Another disadvantage with this type is the gradual loss of active material through the open slits by way of which the electrolyte gains access to the active material.

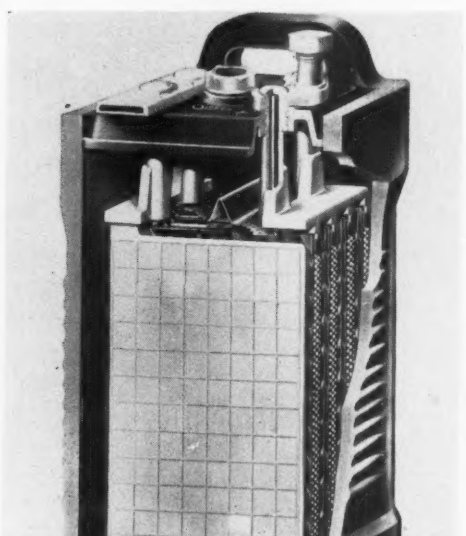
Characteristics of the "Pg" battery

The manufacturers, Oldham & Son Ltd., Denton, Manchester, have now introduced a plate of new construction, termed the "Pg" as an abbreviation for "polyvinyl chloride-glass fibre". It is a double-sleeve, multi-tube positive plate which combines the benefits of vitreous felt retainer mats and of plastic outer insulation. This type of tubular positive plate resolves some of the problems which have been the subject of research for a considerable time. All the components are virtually immune from the effects of continuous immersion in acid. The mechanical construction is such that shedding of the active material, either by the normal cycling of the battery or by externally excited vibration, is eliminated.

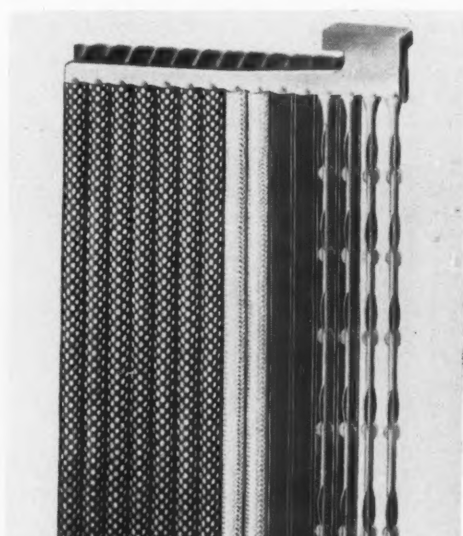
Constructional features

The plate consists of a number of thin P.V.C. tubes each of which, by reason of its greater inherent strength, can be perforated to provide for the electrolyte a greater area of access to the active material than is possible with the con-

Part sectioned arrangement of the Oldham "Pg" battery, showing the plate assembly seated low in the container



Constructional features of the positive plate. Right to left: supporting spine, active material, glass fibre sleeve, P.V.C. outer tube



ventional slit-tube construction. Each P.V.C. tube is lined with a woven glass fibre sleeve, the assembly being centred on a feathered "X" alloy spine. The active material is closely packed into the space between the spine and the glass fibre sleeve. It is apparent that by reason of the thinness of the P.V.C. tube and the glass fibre wall, a greater volume of active material can be packed into a tube of a given overall diameter. This is an important factor which contributes to greater storage capacity.

Operational advantages

In use the glass fibre sleeve becomes securely embedded in the active material, which is thereby positively retained, so contributing to considerably longer life. On account of the increased exposure of the active material to the electrolyte, afforded by the perforations in the P.V.C. tube, the high-rate discharge is better than can be expected from a conventional slit-tube positive plate type of battery. The "Pg" method of construction provides a practical answer to the long-standing problem of retaining the active material around the grid structure. Deterioration of the positive plates has been virtually overcome by the new technique so that the potential service life has been appreciably extended, with a corresponding reduction in battery costs per annum. Moreover, the "Pg" battery requires much less frequent topping-up. As a deep space at the base of the battery is no longer required for sediment, the plate groups are positioned lower in the cell, thus allowing the height of free acid above the tops of the plates to be considerably increased without adding to overall dimensions. Compared with a standard battery of an equivalent performance, a "Pg" traction battery can show an appreciable saving in weight.

The "Pg" battery is conservatively rated. Compared size for size with a standard battery the greater volume of active material provides added capacity, as much as 15 to 18 per cent in traction batteries. This characteristic prolongs service life, since it affords a greater reserve of power. The construction confers the additional advantage of a low rate of self-discharge.

For electric vehicles and mechanical handling

Used in works trucks, fork-lift trucks and delivery vehicles, such as milk and bread vans, the "Pg" battery has the advantage of extra reserve capacity within a given space. This enables more work to be done in a shift or, in the case of delivery vehicles, an extension of a round.

The "Pg" battery has an appreciably extended service life and it requires much less topping up, thus making its contribution to the reduction of maintenance costs. A set of "Pg" bus or coach batteries, as against a comparable set of conventional batteries, can show a considerable saving in weight. Particularly in seasonal coach operation, service life is often prematurely shortened by self-discharge where vehicles are laid up during out-of-season periods. The "Pg" battery with a lower self-discharge rate has obvious advantages for service under such conditions.

SPARK PLUG ASSEMBLY

THE accompanying illustration shows a new, air-operated machine recently developed by the Crusota Engineering Company to speed up the assembly of spark plugs. It incorporates standard Mead air units, and by an ingenious combination of electrical and pneumatic devices drives home the locking nut on a Duchess spark plug in remarkably short time and with a minimum of effort. The operation sequence is as follows:—

(1) The spark plug is placed in the LS-1 collet fixture C.

(2) The main air valve is opened to supply collet pressure. This valve is not shown in the illustration; it is on the far side of the machine.

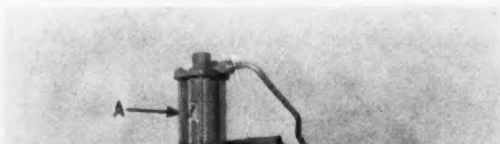
(3) A micro-switch on the left is depressed to make momentary contact and energize a four-way solenoid valve. This causes the double-acting air cylinder A to move the dovetail slide down to carry the nut runner towards the work.

(4) Before the nut runner engages over the hexagon nut of the plug, the cam finger B attached to the dovetail slide trips a micro-switch roller that energizes a two-way solenoid. This admits air to the nut runner. Tripping this micro-switch also starts the timer motor.

(5) A built-in clutch controls the nut runner socket, and the timer is adjusted to allow a supply of air that is just sufficient to throw the clutch. When the clutch is thrown, the solenoid controlling the air supply to the nut runner is reversed by the timer, thus releasing the contact pressure between the socket and the nut, ready for upward travel of the socket.

(6) Depression of a micro-switch for momentary contact, shifts the four-way solenoid valve, and the double acting cylinder moves the dovetail slide up to the start position. The upward travel of the slide removes the cam finger from the micro-switch to de-energize the timer motor ready for re-setting for the next operation.

The complete time cycle on this machine is three seconds.



This American machine automatically drives and torques a spark plug lock nut in three seconds

PETTERS PC ENGINES

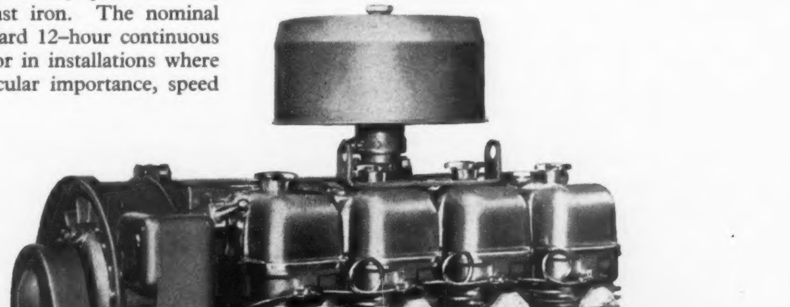
A Range of Small-capacity, Air-cooled, High-speed Diesel Units having a Wide Field of Application

WHILE the small diesel engine has established its suitability for industrial and stationary duties, its incursion into mobile, traction and automotive fields has been retarded by considerations of its relatively low speed of operation, higher specific weight, and greater bulk. This new range of air-cooled units introduced by Petters Ltd., of Causeway Works, Staines, Middlesex, is intended to overcome, or at least to substantially reduce, these handicaps. The aim is at producing engines that, basically, may be classed as "universal" power units. Embodying standardized component parts throughout the range, they will be available in stationary, portable, marine, traction, and automotive versions. Prototypes have been in the hands of operators for test under normal service conditions for the past two years, and extensive production facilities are now being organized at the Staines plant.

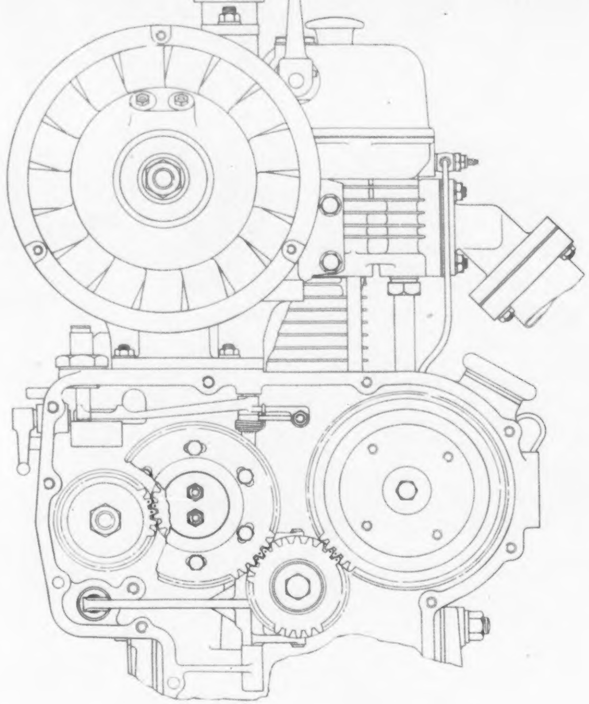
Concurrently with the aims of the general design is an insistence on operational reliability and an ability to withstand inexpert handling under arduous working conditions in exposed locations in any climate. To this end, the engines are very conservatively rated and the construction may be termed "rugged". In spite of the desire to save weight, these considerations are given precedence and no recourse is had to light alloys for structural parts. Cylinders, cylinder heads, crankcase, and sump are all of cast iron. The nominal output in all cases is a British Standard 12–hour continuous rating. For automotive purposes, or in installations where the power : weight ratio is of particular importance, speed and output may be raised and the engine specially rated on a British Standard 1–hour continuous basis.

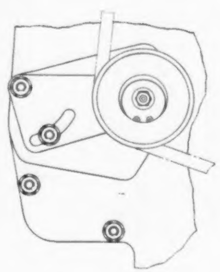
The range includes one-, two-, three- and four-cylinder units having common cylinder dimensions of 3 in (76·2 mm) bore and 3 in (76·2 mm) stroke, giving a swept capacity of 347·38 cm^3 per cylinder and 1,390 cm^3 for the four-cylinder unit. At the nominal rating, the output per cylinder at the relatively low b.m.e.p. of 62 lb/in^2 is 5·0 b.h.p. This is attained at a rotational speed of 3,000 rev/min, equivalent to a piston speed of 1,500 ft/min. Fuel consumption at full load at 3,000 rev/min is 0·52 lb/b.h.p/hr. The specific weights of the dry engines, complete with flywheel, at their rated outputs are 48·6, 32·9, 29·2 and 25·1 lb/b.h.p. for single-cylinder to four-cylinder units respectively.

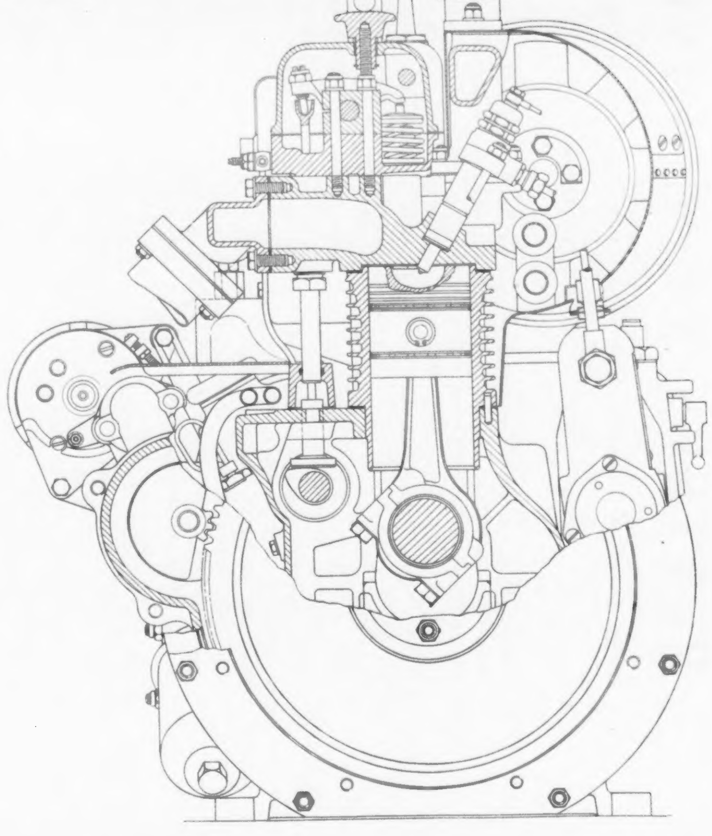
In two major aspects, the single-cylinder engine differs from the multi-cylinder units. It has a tunnel crankcase into which the crankshaft is threaded from one end and access to the connecting rod big end is through a side door. For cooling, a flywheel type fan is employed. The multi-cylinder units have an open crankcase, fitted with a shallow sump, and the crankshaft is underslung in the usual manner. Cooling is by means of a belt-driven axial-flow fan mounted



Petters PC4 four-cylinder air-cooled diesel engine. Industrial version developing 20 b.h.p. at 3,000 rev/min on B.S. 12-hour rating. Fitted with SAE No. 5 flywheel housing, shaft extension, and electric starting



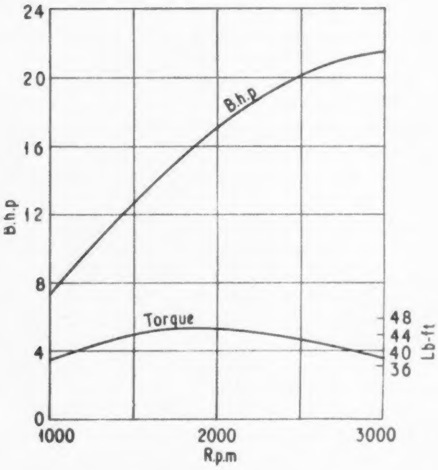




PETTERS PC DIESEL ENGINE

Bore 3 in   Stroke 3 in

Output (12-hour rating) 5 b.h.p. per cylinder at 3,000 rev/min. Two-, three-, and four-cylinder units are air-cooled by a standardized axial-flow fan belt-driven at different speeds



Power and torque curves for the PC four-cylinder engine. At full load and 3,000 rev/min the fuel consumption rate is 0·52 lb/b.h.p./hr

at high level and delivering directly into a duct alongside the cylinder heads.

The deep-section crankcase is of cast iron and is heavily ribbed for rigidity. Crankshafts are machined from En 19 alloy steel stampings and are induction hardened and ground on the journals and crankpins. Each throw of the shaft is supported between two, tin-flashed, copper-lead lined, thin-walled, precision main bearings and end thrust is taken by copper-lead lined washers. All shafts are statically and dynamically balanced. As regards the cast iron flywheel, the method of attachment to the shaft differs on single- and multi-cylinder units. For the single, the wheel is located on a taper and keyed in position, but on each of the multi-cylinder engines it is registered and bolted to a flange on the shaft. A flywheel housing to suit one of a range of S.A.E. standard bellhousings is supplied as standard equipment. To meet installation requirements, the flywheel housing is provided with either feet or side attachment pads.

The finned individual cylinder barrel is a simple casting in high grade iron and has a wall of a section adequate to permit reboring, when necessary, to accommodate oversize pistons. It is deeply spigoted in the crankcase and, since crescented cut-outs are machined in the upper end of the barrel to furnish clearance for the valve heads, the base flange of the barrel is dowelled to the crankcase to provide a positive angular location.

The cast iron individual cylinder head, presenting a recessed flat face to the working space, is located on the upper spigot of the barrel and seated, with an interposed gasket, on a substantial circular flange. Porting is arranged laterally, with the exhaust to the side of the engine opposite to that of the incoming air.

Valves of En 59 steel are mounted vertically and, being to one side of the vertical, longitudinal plane at the cylinder axes, the valve heads slightly overlap the bore of the cylinder, which is relieved locally for clearance. The inlet valve, of a diameter larger than the exhaust valve, is shrouded to promote a swirling motion of the air entering the cylinder and, of course, is non-rotatable. A split pin positions on the valve stem the spring retainer, which has a slotted peripheral lug slidably engaged on a hardened vertical guide pin. Each valve is fitted with two concentric springs seated deeply in the shallow valve casing. This casing is attached to the cylinder head by two long studs which also serve to clamp the short spindle for the rocker levers in its support block. One of these studs is extended to receive the hand nut that draws down the light alloy rocker enclosure. Each assembly of cylinder and cylinder head is secured to the crankcase by four long studs, enabling any head or barrel to be removed without disturbing the other units of a multi-cylinder engine.

En 8Q alloy steel is used for the H-section connecting rod, the big-end of which is divided at 45 deg so that, if necessary, the rod can be drawn through the crankcase aperture. It will be noted that the big-end joint face is rabbeted to relieve the set bolts of shear stress. The big-end bearing is of the copper-lead lined, thin wall, precision type and the small-end bush is also of lead bronze.

In the flat face of the Lo-ex alloy piston is formed an axially-located, approximately hemispherical, combustion cavity and also two shallow recesses to furnish clearance for the valve heads. Circlips position and retain the fully-floating gudgeon pin. The ring assembly comprises three compression rings and two channelled oil-control rings. While the top ring is chromium plated to reduce wear in the bore, the second and third rings are taper faced. Both the oil rings are chromium faced and the upper one, above the gudgeon pin, is of the split type to secure the optimum scraping action.

Valves are operated by individually enclosed push rods actuated by flat-base tappets from a camshaft mounted high in the crankcase. The camshaft, of carbon steel with hardened cam profiles and journals, runs in pressure-lubricated, white-metalled bearings and is driven from the crankshaft by a train of hardened helical spur wheels at the forward end of the engine. Valve clearance is adjustable by a screw in the end of the overhead rocker lever engaging the cupped end of the push rod. Push rod enclosure tubes are secured to the cylinder head and are sealed at their lower ends in a tappet-head cover bolted to the face of the crankcase. An external pipe from the mainshaft gallery in the crankcase conducts oil under pressure to the valve casings and thence by drillings to the overhead rocker levers. The supply of oil to each casing is adjustable by means of a needle valve. Inlet valves are timed to open 10 deg before T.D.C. and close 50 deg after B.D.C., while the exhaust valves open 45 deg before B.D.C., and close 15 deg after T.D.C.

Bryce-Berger fuel injection equipment is employed. On the single-cylinder engine the injection pump is mounted on the crankcase and operated from a cam on the engine cam-

The Lo-ex piston has three compression and two scraper rings. Top compression ring and the two oil rings are chromium plated

Individual cylinder heads are standardized throughout the range of units. Intake and exhaust are ported to opposite sides of the head

shaft by a tappet and push rod. All multi-cylinder engines, however, are fitted with a self-contained pump unit flange-mounted on the rear of the timing case and driven from the timing gear train. A centrifugal governor is mounted on the engine crankshaft behind the half-speed pinion and the linkage to the injection pump rack control is completely enclosed. Stressed elements of the linkage are in direct line with governor fly-weights and the connection to the pump mechanism at a higher level is consequently of light section.

Engine speed is controlled within the limits stipulated by B.S. 649, and speed may be varied 10 per cent higher or lower by means of an externally adjustable abutment. A full-range speed control is, of course, fitted for engines intended for automotive applications. A spring-loaded overload stop, which is automatically reset after operation, is also provided. The spill timing of the injection pump is specified at 29 deg before T.D.C., and fine timing of the pump is effected by angular adjustment of an intermediate driving wheel on its flanged hub, as may be seen on the assembly drawing.

A common assembly of nozzle and nozzle holder is used for all the engines. The holder is mounted in the head at a transverse angle of 30 deg from the vertical and with the two-hole nozzle tip positioned on the cylinder axis. The two fuel sprays diverge in a longitudinal plane and all fuel is directed into the piston cavity. Fuel is passed through a Cooper filter of the renewable paper element type and is lifted from the tank by an AC diaphragm pump mounted on the injection pump casing and operated from the pump camshaft. Air for combustion is drawn through a large diameter cleaner, also of the paper element type, mounted vertically on a distributing manifold enclosed in the cooling air ducting.

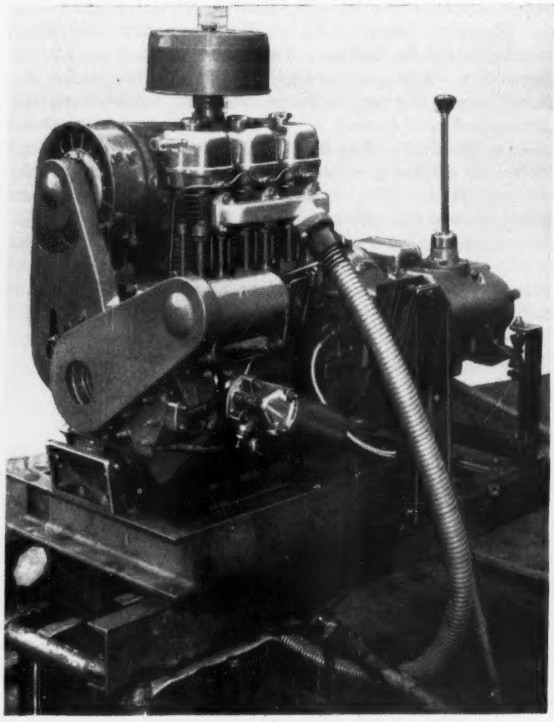
Lubricating oil is circulated under pressure by a gear pump mounted on the outside of the timing cover and gear-driven from an intermediate pinion of the timing train. The oil is filtered in a Purolator paper element unit mounted externally on the crankcase. Using the standard shallow sump, the engines can be operated at various angles of tilt, either fore and aft, or radially. Longitudinally, for two-, three-, and four-cylinder units in that order, the permissible tilt is 20 deg, 10 deg, and 5 deg down at the timing end and 20 deg, 12 deg, and 7 deg down at the flywheel end. Radially, all three units can be tilted 30 deg clockwise and 15 deg anti-clockwise as viewed from the flywheel end. A specially deep alternative sump may be fitted where angles of tilt in excess of the foregoing are necessary. For certain duties, mostly industrial applications, an oil cooler may be required. A tubular cooler unit may be mounted horizontally inside the main cooling air trunk.

For the relatively small amount of cooling air required for the single-cylinder engine, the conventional flywheel impeller is adequate and was adopted for its constructional simplicity. The multi-cylinder units, however, could be better served by a small separate fan operating at a speed considerably higher than that of the engine crankshaft. A decision was taken to use an axial-flow fan, rather than a centrifugal fan, as the axial type can be more compactly installed and requires a neater and simpler ducting and cowling arrangement. The cooling effect of air is proportional to its rate of flow and is largely independent of its pressure. Fan diameter is influenced by the physical dimensions of the engine to which it is fitted and the maximum speed of the impeller is limited by the blade tip speed at which the noise level becomes obtrusive. High overall efficiency is essential, or too large a proportion of engine power is absorbed in driving.

For the PC engines, a high-solidity axial fan with inlet guide vanes and an 8 in diameter, eight-bladed impeller was designed and developed, and eventually standardized for the multi-cylinder units. The use of a common fan is an obvious production economy and only variation of the driving arrangements is necessary to obtain rotational speeds and consequent flow rates to meet the requirements of two-, three- and four-cylinder engines. Twin belts transmit the drive from the crankshaft to the fan, with a tensioning pulley mounted on an adjustable, pivoted bracket on the timing cover. The impeller spindle is mounted in a roller bearing and a ball bearing.

In the case of the four-cylinder engine, the ratio of the drive is 1 : 1·79, giving a fan speed of 5,375 rev/min at the rated maximum engine speed of 3,000 rev/min. At that speed, the fan static head is 1·2 in water, the delivery of air is at the rate of 934 ft^3/min, and the power absorbed in driving is 1·15 h.p. The main delivery duct from the fan is

A Petters PC3 air-cooled diesel engine powers the new Mercury ACD industrial tractor which has a drawbar pull of 2,000 lb



Suspension of a three-cylinder traction unit on Metalastik sandwich-type mountings. Torque-reaction abutments are fitted on each side of the gear unit

arranged alongside the cylinder heads and encloses the combustion air manifold, the fuel injectors and, when fitted, the lubricating oil cooler. Access to the injectors for servicing is by way of a full-length panel in the duct. This panel is completely detachable and is secured by Dzus rapid-action fasteners. It may be necessary in certain installations for the cooling air, heated in its passage over the cylinder heads and cylinder barrels, to be ducted away from the engine. Suitable adaptors can be supplied for that purpose.

Since the engines are of the direct-injection type and operate at a compression ratio of 18·5 : 1, starting is positive and easy at normal atmospheric temperatures. Single-cylinder and twin-cylinder units are started by hand-cranking, but 12 V electric starter motors are standard equipment on the three-cylinder and four-cylinder engines. The Lucas motor, which has a solenoid-operated, positive-engagement pinion, is mounted high on the crankcase on the exhaust side of the engine and is shielded from the heated cooling air leaving the cylinders by a horizontally arranged deflector plate.

Although the engines can be started on full compression, decompressor mechanism is fitted to make hand-cranking easier and to facilitate servicing operations. The decompressor gear comprises a throw-over cam, which engages an abutment lug formed on the rocker lever for the exhaust valve. As the cylinder heads are individually demountable, short decompressor camshafts are connected by means of Oldham couplings for operation from a single control lever. The loose elements of the couplings are pinned to their respective shafts to prevent accidental loss when a head is lifted.

The dynamo fitted as standard is the 12 V Lucas C.39P2 unit, belt-driven at 1·53 times engine speed. Maximum current generated is 16·5 A, giving a capacity of 198 W. The cutting-in speed of the dynamo is 2,000 rev/min, which is approximately equivalent to an engine speed of 1,300 rev/min.

As standard, the direction of engine rotation, viewed from the flywheel end, is clockwise on single- and twin-cylinders units, and anti-clockwise on the three- and four-cylinder engines. The design of the engines, however, makes it possible for these rotational directions to be reversed if necessary.

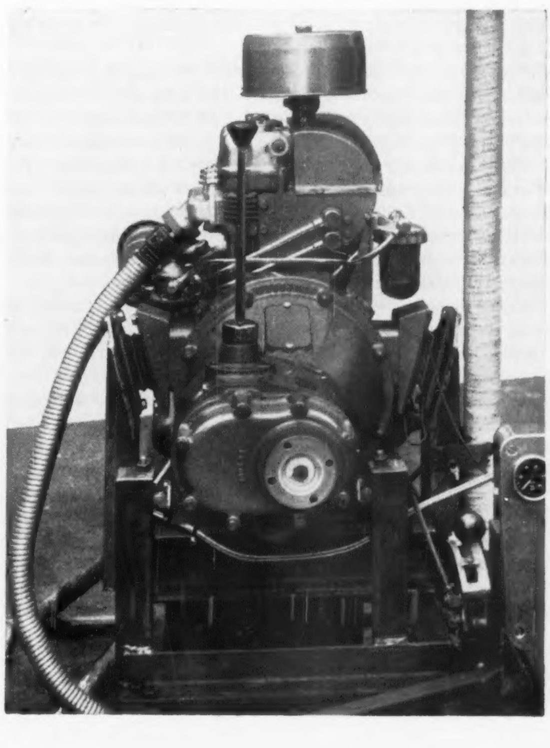
To meet the requirements of a wide variety of applications, an extensive range of suitable auxiliary equipment is available. Over-centre, plate-type clutches and 2 : 1 and 4 : 1 reduction gears can be fitted for industrial or marine engines. Traction units are equipped with heavy-duty, three-speed and reverse gearboxes by P. R. Motors Ltd., and Borg and Beck clutches. For automotive units, the Standard small car gearbox with four ratios from 1 : 1 to 4·27 : 1 and reverse is used for the twin-cylinder engine. The Morris N.G.L. gearbox with four ratios from 1 : 1 to 4·99 : 1 and reverse can be fitted to the three- and four-cylinder units.

As in many industrial applications the engines will be required to operate continuously under adverse atmospheric conditions, special care has been taken to provide for the efficient cleaning of combustion air and filtration of the fuel and lubricating oil. All working parts are enclosed and the unit generally is dust-proof. The only exceptions are the belt drives for the cooling fan and the dynamo. Engines for industrial or other exposed installations are equipped with belt shields, but these shields are not required in bonnetted automotive installations.

The illustrations of a three-cylinder traction unit show a resilient three-point mounting system using Metalastik components. Interleaved, multiple rubber sandwich mountings arranged at an inclination to the vertical are positioned low at the centre front and high at each side of the flywheel housing. Rubber torque-reaction abutments are provided on each side at the rear extremity of the gear unit.

Overall dimensions of the engine are: height, including air

Rear view of a traction unit fitted with a P.R. Motors three-speed and reverse gearbox. The connections to a lubricating-oil cooler located in the main air duct will be seen above the bell housing



cleaner, 30⅞ in; width, over cooling fan and dynamo, 21¼ in; and length, including belt shields and flywheel housing, 23 in for the three-cylinder engine and 28¼ in for the four-cylinder unit. Throughout the engines Unified screw threads are used and, wherever practical, Simmonds Pinnacle self-locking nuts are fitted. Conforming to international practice, screw threads on the fuel injection equipment are to metric standards.

Petters Ltd. have an exceptionally long record in the production of small internal combustion engines and the Company's recent experience relating to methods of cooling is of some interest. Three years ago their sales of small units were in the ratio of 3 : 1 in favour of water cooling. Currently, however, the trend has been reversed and sales are now proportioned at approximately 3 : 1 with preference for air cooling. The small industrial tractor illustrated has recently been introduced by the Mercury Truck and Tractor Co. Ltd. of Gloucester. It is powered by the Petters three-cylinder air-cooled diesel engine combined with an automotive type three-speed gearbox and has a low-ratio worm-drive transmission to give a drawbar pull of 2,000 lb. Designated Model 20 ACD, it is the diesel-engined version of the Mercury Model 10F petrol-engined tractor which has been widely used in industry and by railways, air lines, and public authorities during the past fifteen years. Although the overall dimensions of the tractor are only 78 in long and 48 in wide, it is capable of moving loads up to 15 tons on a level surface.

Transfer Pressing*

The Application of Automation to Press Work

THE tremendous advantages to be gained from automatic production in the press shop are clearly illustrated by what happened when a transfer press was introduced to make a certain part for a car engine. Before the advent of the automatic machine, six, 84 in, 300 ton straight-sided presses were used to produce this part. With the introduction of the transfer press, it became possible to make the complete part on one machine.

The benefits to be gained from such an innovation are not limited to greatly increased production, with subsequent cost reduction, although this is very marked. Saving in floor space can be considerable, and in this particular instance amounted to some 50 per cent. Furthermore, the savings in materials handling was estimated to be 75 per cent.

But one of the most important developments resulting from the introduction into a factory of such manufacturing aids as transfer presses—indeed, perhaps the most important—is the fact that each worker's "zone of safety" is inevitably increased. With automatic controls, it becomes unnecessary for a worker to approach a machine in motion.

A proper approach to efficient press production must take note of several important factors. Not only must means be provided for a high rate of production with a minimum of manpower, but means must also be provided to reduce to an absolute minimum the change-over time from the production of one type of stamping to another. This change-over must be accomplished in minutes rather than in hours. Furthermore, automatic press equipment must be flexible, that is, it must be able to produce a variety of stampings to-day, and a completely new or redesigned product tomorrow.

These three concepts—automatic transfer and handling of work; rapid change-over time; and complete flexibility and versatility—have been very much borne in mind by the Clearing Machine Corporation in designing and building their latest equipment. They have not only dictated new mechanical designs for presses and press equipment but, equally important, they indicate a completely new approach to press plant design and operation.

Automatic transfer and handling of stampings

It has long been recognized that transfer type presses are singularly adapted to efficient high speed production, and in fact these presses have been used in Europe for some time, although only for the production of small components. Transfer presses are designed to perform numerous operations on a component and thus produce a completed part within one press. In this type of press, the part is transferred from station to station by a feed mechanism that is mechanically driven from the press itself.

Synchronization of the press and feed mechanism is automatic and consistent. There are no limit switches to be tripped, no relays to operate, no air valves to actuate and no air cylinders which must be timed. Compare this with the typical automatic press line, which requires that dozens of limit switches, relays, air valves and cylinders be operated every time a part is produced. Malfunction of any device puts the press line out of operation until repairs or adjustments are made. Another shortcoming of this type of line automation is that the part is pushed, pulled, shoved, flipped and conveyorized, which means that control of the work is often lost between each press and relocation is necessary.

Transfer press automation

Transfer press type of automation consists in very definite right-angle motions, which give positive and accurate indexing of the work. That the part is always under positive control is the reason for the greater efficiency of this type of automation. The basic concept of press line automation held by the Clearing Machine Corporation is that a line of presses is essentially one big transfer press, with only the station centres increased.

The major difference between the Corporation's first approach to this type of automation and the present practice is that only one mechanical source is used to power the automation for a complete line of presses, when the line is less than 75 ft long. The mechanical source is similar to the feed operating mechanism of the Transflex press. For line automation, the feed operating mechanism is driven continuously by a variable speed motor through a clutch and brake unit. The presses are single-stroked by a feed mechanism. In this way the speed of the press line can be changed and the presses and feed inched independently.

The conventional transfer press automation consists of two motions—a clamping and unclamping motion, and an index or transfer motion. This has always limited the transfer press automation to parts with a comparatively flat flange or edge. A recent Clearing development introduces a third, or "lift", motion to these mechanisms. It is now possible to use this type of automation with virtually any type of stamping. It is adaptable to both the transfer and to the automatic press line. A single source is used to power all the automation, including a turnover for a complete line of presses. Only one limit switch is required for the press line synchronization. This

**Based on a paper presented by Gordon M. Sommer and Robert H. Barlow of the Clearing Machine Corporation to the Inst. Prod. Eng. Automatic Production Conference.*

means that for the first time automation of complete lines of presses can be as precise and efficient as a transfer press.

In addition to having automatic equipment that is precise and efficient, it is necessary to consider change-over time. Clearing Machine Corporation have recently developed "push button" die setting, so that the change-over can be reduced from hours to minutes. This development makes it entirely feasible to change a press or press line from one part to another every four to eight hours.

A great deal is said these days about the advantages of integrated production. Integrated production means that the parts are produced only as required, thus eliminating large stockpiles and inventories. Since it is obviously not feasible to have a separate production line, or transfer press, for every type of stamping, it is equally obvious that press lines and transfer presses must be equipped with means for quick die and automation changes, if integrated production is to become a reality.

"Push Button" die setting

With "push button" die setting and Transflex automation, change-over time is reduced to a minimum. The old dies are removed from and the new ones secured in the press automatically and by push button control. Since the dies are loaded at right-angles to the work flow, and since Transflex automation consists of straight line motions, the change-over consists merely in replacing sets of feed channels. These channels, with the appropriate feed fingers, become a part of the tooling set-up and are, in essence, a part of the die equipment.

A press so equipped has two, specially-designed bolster structures. They are mechanically connected and are power-driven to the right or left through the press uprights. While a production run is in progress, the bolster not in use can be loaded with the die for the next production run. The bolster is designed so that standard pressure pins can be placed outside the die space area and the lower die shoe completely secured. After the production run is completed, the bolsters are power-driven to the left. This removes the die that has been used and places the new die in the die space area. The old die is now at the left of the press. It can be removed at any convenient time and replaced by another die for another production run, for which the bolsters will be moved to the right. Dies can be changed in complete lines of presses simultaneously, without any waiting for overhead cranes or die trucks. The punches are clamped to the slide, or slides, by means of automatic or semi-automatic clamps.

Since a basic principle of "push button" die setting is to load the dies at right-angles to the flow of work, the moving bolsters are operated front to back in a Transflex press. The advantage of this arrangement is that an absolute minimum of automation is removed for die setting. Two sets of feed tubes and channels are mounted on the bolsters, so that the feed fingers, as well as the dies, can be mounted and adjusted during press operation. The feed tubes can be readily coupled together during the change-over time.

One of the latest Clearing developments is a Transflex press for the production of two different components for two different products. It has an overall length of 58 ft, five slides, a transfer feed stroke of 40 in, and will produce more than 700 stampings per hour. This press incorporates many advanced concepts of flexible automation. These were not all envisaged in the initial stages of development. In fact, some were evolved to meet the specific production problem.

The original request was for a transfer feed press which could handle the seven operations required for either part. It was intended that dies should be changed whenever production was to be shifted from one product to the other. Under these conditions, runs would have to be relatively long, and inventories would have to be maintained on both types of parts. Moreover, the die change-over would have taken considerable time out of the production schedule. It should be noted that moving bolsters had not then been fully developed.

The first suggested improvement involved a reconsideration of die design in order to use some of the dies for both parts. This would lessen the number of dies to be changed and therefore shorten down time for change-over. Clearing development engineers carried this a step further and suggested a press that could hold all of the dies required for both parts, and in which change-over could be effected by slide adjustments. This would eliminate the manual work involved and would shorten change-over time from hours to minutes.

In addition to the more obvious advantages of reduced labour costs and less down time, this innovation made possible completely new thinking with regard to production scheduling. Change-over could be more frequent and inventories kept to the very minimum. Thus, production could be closely integrated with the assembly operations for the two products. In point of fact, the change-over is made in 10 minutes.

Putting both sets of dies in the press at the same time increased the size of the press from seven to nine stations. It also created some problems of unbalanced loading. However, multiple-slide arrangements overcame these. The addition of several between-slide uprights further increased the size of the press. Split slides were then conceived to eliminate some of the extra uprights and so reduce overall dimensions. Split slides are a pair of slides which operate between a single set of uprights, and guide against each other at the middle. The additional benefits of split slides—individual die adjustment and more satisfactory performance under conditions of unbalanced loading—also proved advantageous in the press.

Even after the removal of several uprights through the use of split slides, the press was still remarkably large. It now measures over 45 ft left to right across the main frame structure. It is doubtful whether a press of these dimensions could be constructed in a conventional manner. Certainly, shipment and erection in the customer's plant would have been difficult, if not impossible. Furthermore, a structure of this size could be a monstrous white elephant if production requirements changed radically.

Modular construction

Because of this the Clearing Machine Tool Corporation developed the concept of modular construction. This particular press breaks down into two basic units, that is, two crowns and two beds are provided. They are joined together by a common upright to form one structure. Individual drive is provided for each crown, but the two drives are mechanically coupled together. The press is practical to manufacture, easy to erect and easy to ship. Furthermore, if more die stations are needed in the future, the modular frame components can be taken apart, additional components added and the whole reassembled into a larger machine. Similarly, if the big machine is no longer required, the common upright can be replaced by two standard end uprights and the result will be two independent smaller presses. This is highly integrated production at its best.

With the increasing use of automatic equipment, it became apparent that concern was being felt regarding the effect on the future of industrial workers. If these wonderful new machines could each do the work of several of the old type, and need only the minimum of attention, what would happen to the men displaced by such a development? Time has shown that such anxieties are without any real foundation. In many companies the introduction of the presses described and other similar equipment has so increased production, enabling the manufacturers to sell to bigger markets, that plants have had to be enlarged to meet the demands. Consequently, the men displaced initially have merely been transferred to other sections and processes.

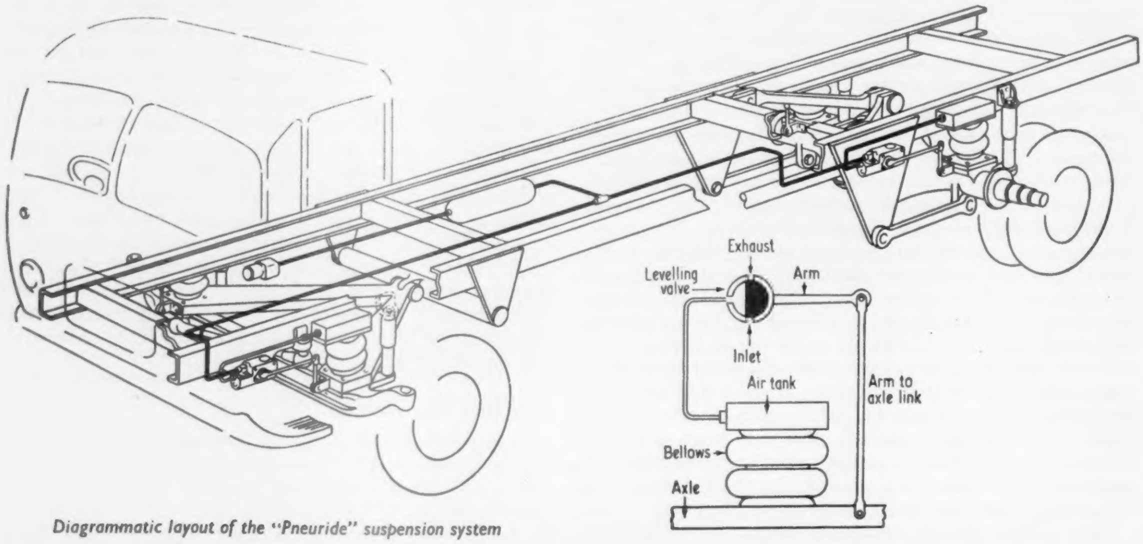
PNEUMATIC SUSPENSION

The "Pneuride" System Developed by Dunlop Rubber Co. Ltd.

AIR as a suspension medium has the outstanding advantage that "spring stiffness" can be controlled to increase with deflection instead of remaining constant as with conventional forms of springing. In the "Pneuride" pneumatic suspension developed by Dunlop Rubber Co. Ltd., for road vehicles, full advantage has been taken of this characteristic to provide a low-rate spring for small movements, while at the same time allowing a higher rate for larger deflections. As a consequence the build-up of resonant oscillations in the system is avoided.

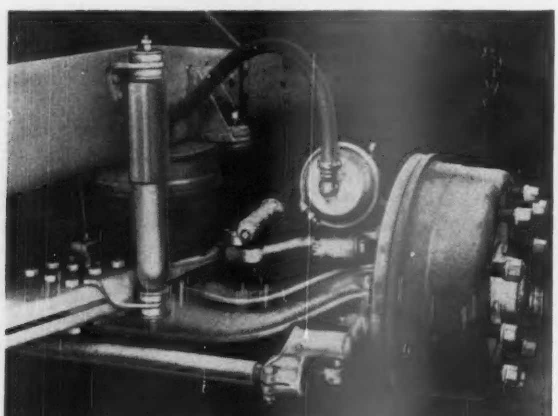
In place of the normal steel springs, the "Pneuride" suspension uses flexible air springs of convoluted-bellows formation in conjunction with a levelling valve that automatically compensates for variations in vehicle loading. Conventional dampers are retained. The air spring is constructed of rubber and nylon cord under methods proved by long experience in the tyre industry, while the levelling valve has been developed specifically to meet the requirements of the pneumatic system.

As is well-known, in some forms of suspension dry friction renders the spring almost inoperative under light loads, with the result that the vehicle is cushioned mainly on its tyres. In contradistinction, because there is no inherent friction in the "Pneuride" air spring, constant performance is assured under all weathers and the shock absorbers can be specified for given conditions. Some damping is contributed by the interflow of air between the air spring and its associated air tank, thus reducing the amount of work that the

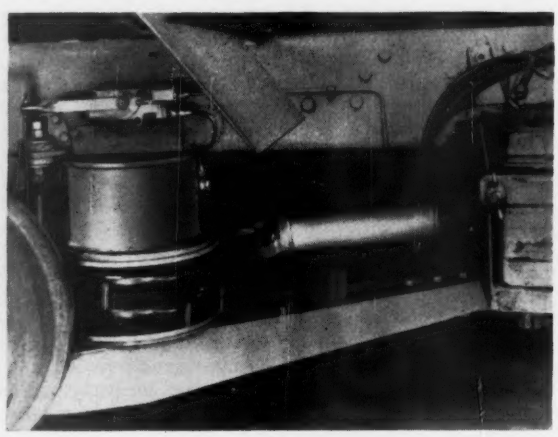


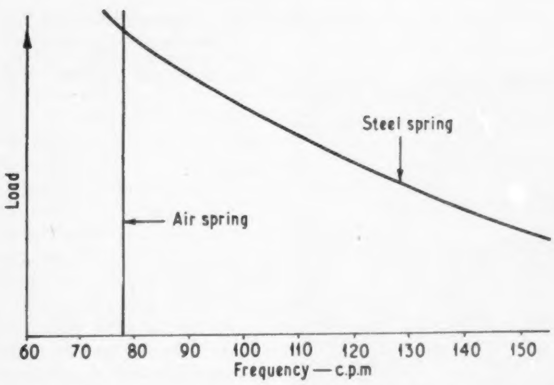
Diagrammatic layout of the "Pneuride" suspension system

Air spring fitted to a front suspension system

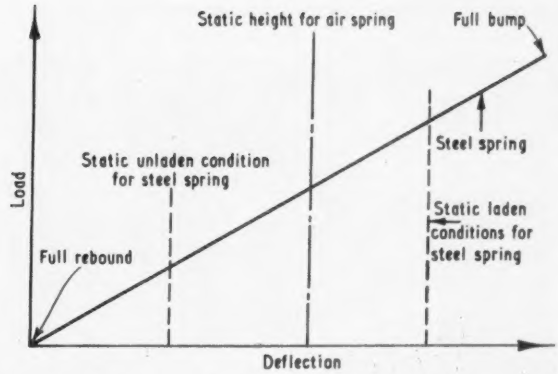


Air spring fitted to a rear suspension system





The air spring system has a low frequency for all loads. A steel spring has a very high frequency with light loads



For all loads, the static deflection of the air spring system is constant. The static deflection of a steel spring varies with load

shock absorbers are called upon to do in service.

A major problem with heavy vehicles has hitherto been the high-frequency vibration set up when they were used unladen or when carrying bulky loads of relatively low weight, because the springing was generally designed to cater for maximum load conditions and was, therefore, too stiff for low load conditions. In the "Pneuride" suspension system, whether the air springs are lightly or heavily loaded, the air pressure is automatically adjusted by the levelling valve. This ensures maintenance of the designed frequency to give a similar ride under all loadings. The standing height of the chassis relative to the wheels remains constant.

In addition, full spring travel is assured by the control exercised by the levelling valve, and the vehicle does not depend upon bump stops when heavily loaded or upon rebound stops when travelling light. For any loading, the optimum position for the "Pneuride" suspension is approximately the mid point of available travel. A "delay device" incorporated in the valve ensures that normal movements of short duration do not bring it into operation. This delay functions only during valve opening, closure being effective immediately the mid point is reached. Provision has also been made for over-travel, and the valve cannot be damaged by further movement when fully open. The air spring has been designed for long, trouble-free life. Its motion is such that the stresses imposed are exceptionally low so that there is virtually no risk of fatigue failure. Lengthy tests under extreme conditions have shown that the possibility of failure in service can be discounted. However, if for any reason it should be necessary, the air can be released from the system and the vehicle can ride to base on the rubber stops provided.

Each air spring is connected to an air capacity tank. This arrangement increases the volume of air acted upon by the spring deflections and thus reduces the rate of the spring. The tank may be fitted directly to the air spring, or may be located elsewhere and connected by piping. Suitably sealed frame members or radius arms may be adapted as capacity tanks. Normally the maximum working inflation pressure of the air springs is 100 lb/in^2. In practice, pressures are rarely required above 80 lb/in^2, a figure readily provided by the vehicle compressor. For lighter vehicles replenishable containers may be used. The air supply necessary is, in fact, small, since by reason of the delay device air is admitted only when the loading is increased.

In addition to the improved ride, pneumatic suspension also leads to a considerable reduction in noise level, a factor of importance for public service vehicles. It is also claimed that the "Pneuride" air suspension system not only offers increased efficiency, but will also reduce maintenance and replacement costs. Its use in conjunction with rubber-bushed radius arms completely eliminates the need for lubrication. Furthermore, as the system provides a consistently soft and controlled suspension for all conditions, additional loads on the tyres are minimized.

Lubricant for Free-Piston Engines

A SPECIAL lubricant for free-piston engines has been developed by Shell and is now available, where required, throughout the world. Tests leading to the development of this lubricant have taken six years and have been done in collaboration with Société Industrielle Générale de Mécanique Appliquée of Lyon, France, and General Motors in the United States; it is already in use in practically all the gas generators built by SIGMA.

Basically, the free-piston engine is similar to a gas turbine, with this difference—a reciprocating instead of centrifugal blower is used. There are two pistons in a horizontal cylinder and fuel is supplied between them. When the pistons advance, the fuel-air mixture is compressed until it fires and the explosion throws the pistons apart. Compression of air behind them moves them forward again. The hot exhaust gases feed a turbine. It was the combination of the high compression pressures and high heat release in the cylinders which originally caused lubrication problems.

Late in 1950, when SIGMA had begun small-scale production of GS. 34 units, experience had revealed that cylinder and piston ring wear rates and cylinder and piston cleanliness were unsatisfactory.

Various Shell lubricants were tested in the following two years and the best was subjected to a 500-hour endurance test in 1952 in a two-cylinder electric generator unit at Rheims. At the conclusion of this test a strip-down of the engine revealed a marked improvement in cleanliness and a sharp reduction in wear. An additional 600-hour test produced similar results and also a decided decrease in compressor delivery valve deposits and air-box fires, as a result of a joint effort of both the free piston engine maker and the Shell Company. Over the next four years refinements and improvements were made in each successive oil based on the test results of SIGMA engines. Concurrently with the Lyon and Rheims test, long term tests were also being run by General Motors in the United States, on a SIGMA built GS. 34 generator. The result of these tests was the development of Shell Free-Piston Engine Oil.

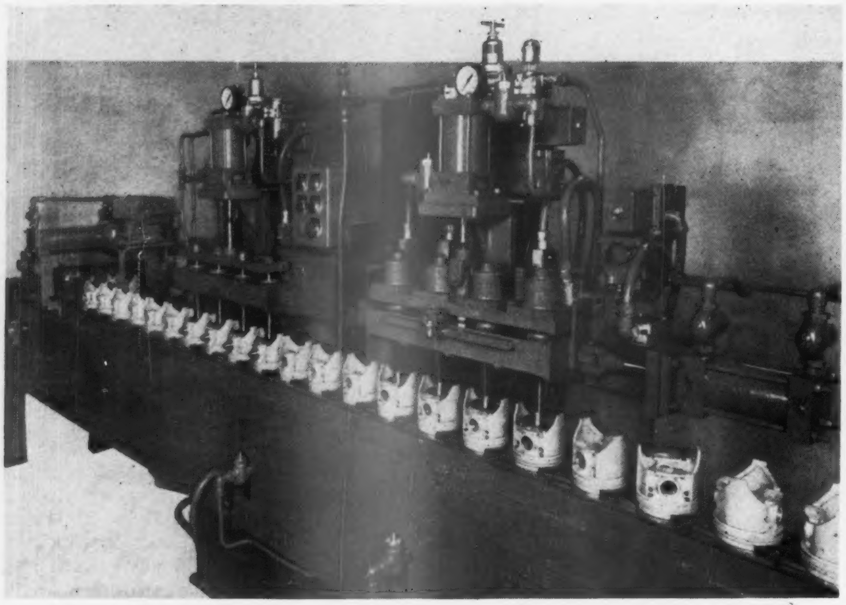


Fig. 1. Automatic bearingizing machine for sizing gudgeon pin holes to within 0·0002 in and imparting a good surface finish

AMERICAN MACHINE TOOL DEVELOPMENTS

Interesting Special-Purpose Machines Recently Introduced in the U.S.A.

In the past few years great attention has been paid to the development of mammoth transfer machines for use in the American automobile industry. The interest in these huge machines has tended to overshadow other developments, which, though smaller and less spectacular, can nevertheless lead to greatly increased output at lower unit cost. A few of the recent American developments in this field are described in these notes.

Automatic bearingizing machine

An automatic machine for sizing and finishing the gudgeon pin holes in automobile pistons is shown in Fig. 1. It has been developed by the Hole Engineering Service of Detroit, Michigan. The machine comprises a conveyor, locating station, bearingizing station and a work-unloading device, all combined to function as a continuous-operation automatic unit. It processes four pistons simultaneously in an eight second cycle, within size accuracy of 0·0002 in and 5 micro-inch surface finish.

Six air cylinders are used to effect automatically all the main functions of the machine. They are electrically interlocked to provide the desired operation sequence, and each cylinder has its own pressure regulator, lubricator, filter and speed control valve. The cylinders operate from a common shop air supply, and the pressure regulators are set at 25-50 lb/in^2. This is just sufficient to allow each machine function to be adequately performed with maximum economy in the use of air. The speed control valves determine the rate of cylinder piston travel, so that even slight shocks to the machine or workpieces are completely eliminated. In addition, each cylinder has built-in adjustable cushions at each end, which provide additional assurance of shock elimination.

The machine selector switch has two positions. With it in one position, the machine will operate through one complete cycle and then stop. This is for convenience in setting up. When the setting up is completed and the machine is ready for a production run, the selector switch is moved to the second position. This allows the machine to operate in continuous non-stop cycles until the stop button is operated.

Pistons are loaded, either manually or automatically, with the head end down in a one-sided nest on the conveyor platens. The gudgeon pin holes may point in any direction. The operation sequence is as follows: Cylinder No. 1, which is inside the conveyor housing, actuates the conveyor mechanism, causing it to advance four pistons and bring them directly under the locating station. Cylinder No. 2 then operates to lower the four-spindle locator and a spindle is lowered into each of the four pistons. During this advance, cylinder No. 3 actuates the rotating mechanism that rotates the spindles to bring the gudgeon pin holes into the correct position, that is, at right angles to the conveyor.

When the pistons are correctly positioned, cylinder No. 2 retracts its piston rod and raises the four-spindle locating assembly clear of the work, while cylinder No. 3 returns to its starting position. At this stage cylinder No. 1 again actuates the conveyor mechanism, this time to bring the four pistons to the bearingizing station. Cylinder No. 4 then comes into action, and as its piston rod moves down it moves an attached four-spindle holding assembly down to the pistons. Each of the four spindles has a rectangular holder attached at the bottom, and as the spindles move down, the holders are automatically inserted, one in each piston, between the pin hole bosses to hold the piston in a floating position to take the thrust of the bearingizing operation.

For the sizing and finishing operation there are four horizontally-mounted tools, one for each workpiece. Each

tool incorporates a number of rollers, cam actuated to impart 200,000 blows per minute to the surface under treatment. The tools are rotated by an electric motor and are fed in by cylinder No. 5. When the holes are finished, the tools are automatically retracted and No. 1 cylinder actuates the conveyor mechanism to move the four pistons to the work unloading device. Actually, of course, every time No. 1 cylinder actuates the conveyor mechanism, a three-fold operation is carried out. That is, the conveyor simultaneously moves four pistons to the locating station, moves four located pistons to the bearingizing station and moves four processed pistons to the unloading device.

When the four finished pistons are moved to the work unloading device, they move in front of a rail unit, which has a sliding member operated by No. 6 cylinder. The operation of this cylinder causes the rail to move the finished pistons to a belt conveyor for transfer to the next manufacturing process.

Automatic wheel rim line

Hautau Engineering Company, 721 Wanda, Ferndale 20, Michigan, who specialize in the development of automation equipment have recently developed a wheel rim line for the Firestone Steel Products Division. This five-station machine shown in Fig. 2, automatically embosses, pierces and chamfers large wheel rims. Three fifty-ton, hydraulic C-frame presses are integrated with a high speed in-line transfer mechanism to provide a compact high-production machine. The rims processed range in size from 12 in to 48 in diameter.

In order to facilitate rapid change-over from the dies for one size of rim to those for another, the presses are arranged for rapid adjustment in three planes and also have a rotary adjustment. The heavy C-frame construction takes the action and reaction forces of operation within itself, thereby eliminating any deflection of the rest of the machine. In action, the C-frame is rapidly advanced on ball ways to the pressing position by an air cylinder. The press ram is powered by an hydraulic booster cylinder and has a 2 in pressing stroke.

The transfer mechanism has both longitudinal and vertical movement. There is 10 in vertical travel on ball ways. It is hydraulically actuated. A three-station travelling carriage transfers the rims when the lifting mechanism is in the down position. Adjustable clamps for securing the various diameter rims are mounted to the lift mechanism at each station. The travelling carriage reciprocates on large V-type rollers.

As already stated, the machine has five stations; a loading station, three working stations and an unloading station. Wheel rims are fed down an incline to the loading station where a mechanical loader elevates them to the loading position. The travelling carriage moves the rim from the loading station to the first working station and at the same time moves the rims in process forward one station. When the rims are in position, the power clamps are automatically actuated at all stations. The rims are then elevated to the pressing position. At station 2 the embossing operation is carried out, the valve stem hole is pierced at station 3 and chamfered at station 4. Approximately 400 rims per hour are produced.

Punched tape machine control

Punched tapes control all the functions of the Barnesdril horizontal, single-spindle drilling, spotfacing and chamfering machine illustrated in Fig. 3. The machine is designed for machining radial holes in cylindrical parts up to 36 in diameter and up to 30 in long, supported vertically on an automatically indexed 40 in diameter rotary table. This particular machine has been tooled to carry out 248 drilling, spotfacing and chamfering operations on aircraft gas turbine compressor housings.

The horizontal spindle is bored to take No. 3 Morse taper, vertically adjustable adaptors. It is, therefore, necessary to use pre-set tooling. The spindle drive is through V-belts from a four-speed 2 h.p. constant horse-power motor. Spindle speeds, as programmed for the best machining practice, are controlled automatically by the punched tapes. The spindle is mounted in a vertically adjustable housing, which can be moved, under the control of the punched tape, over a maximum distance of 30 in, by a 1 h.p. motor and a ball-bearing precision lead screw. The maximum speed of movement is 60 in/min. A Selsyn motor indicates the desired position of the spindle and feeds the information back to the control system. It is claimed that the controls are so accurate that the spindle position can be repeated within 0·001 in.

Indexing of the rotary table is effected by a $1\frac{1}{2}$ h.p. motor under punched tape control. A Selsyn motor, geared to the table feeds position information back to the control panel. The position can be repeated within 0·001 in on a 40 in diameter. The table position is automatically checked after the table is clamped. Once the table is located, it is hydraulically clamped to flat supporting ways under the periphery. More than 12,000 lb of workpieces and fixtures can be mounted on the table without fear of distortion.

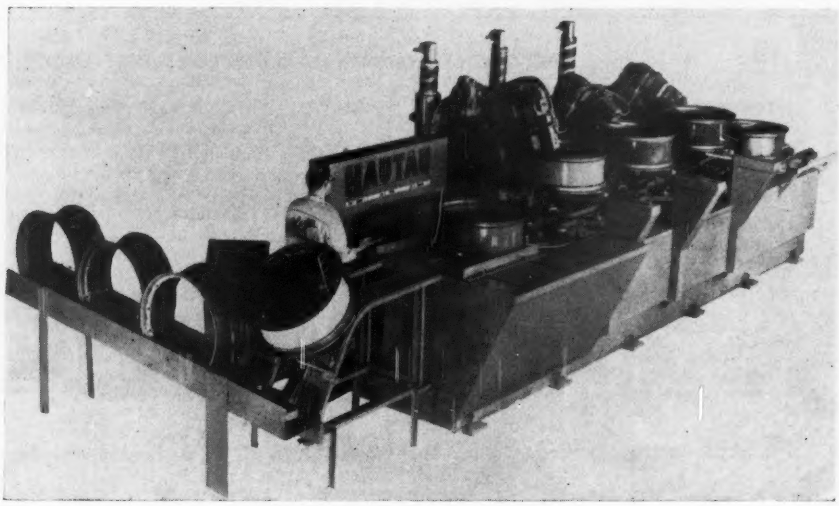


Fig. 2. Hautau, five-station automatic wheel rim line. Rims are embossed, pierced and chamfered, and the line is sufficiently flexible to handle rims from 12 in to 48 in diameter

Fig. 3. Barnesdril single-spindle machine with punched tape control. It is tooled to carry out 248 drilling, spotfacing and chamfering operations

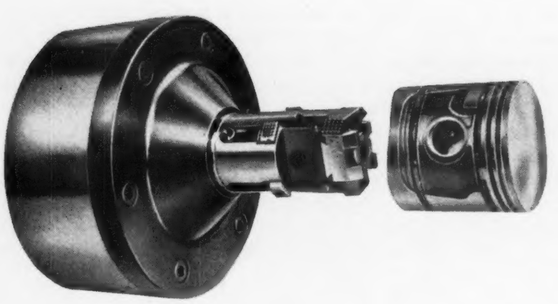
In-and-out movement of the drill head and column is provided by a hydraulic cylinder. It includes rapid approach at 300 in/min and four feed-rates, each of which can be set between 0·5 and 11·0 in/min. Both the spindle and the column feeds are selected as designated by the tape. The controls are so arranged that two feed-rates can be used in sequence, if this is desired. Three sets of cam-actuated limit switches provide three depths of penetration and rapid return as selected by the punched tape. Because the feed is usually to a positive stop, with the depth of cut controlled by pre-setting the tool in its holder, the depth of cut can be controlled within 0·001 in.

The control console houses all the manual control elements and a tool control board. There are positions for holding up to 13 different tools, each pre-set in a holder, in the tool control portion of the console. Any of the 13 positions can be locked out, according to job requirements. The machine will not function unless a tool is placed in each active position on top of the console, except for the tool designated by the tape as the one that should be in use. The control tape can be punched to designate tools in any required sequence. This permits tools to be used in any sequence no matter what their positions may be on the console. Thus, the tool arrangement need not be changed when jobs requiring different combinations of these tools are interchanged to machine a different component.

Tool selection is indicated by the tape. A signal light on the console is activated by the tape to indicate the next tool to use when a step in the machining cycle has been completed. The operator must remove the previously used tool from the machine spindle, replace it in its proper position on the console, and take the indicated tool from the console before the machine can be re-started. For safety reasons, the machine does not restart automatically after a tool change. Each new step in the cycle must be initiated by pressing a push-button on the console.

The control system is built round a numerical positioning control developed specially for this machine by General Electric Co. It uses punched tapes to introduce dimensional control information to the machine. Eight-hole, 1 in wide tapes can be used in any lengths. The holes represent digital dimensions, each transverse row representing one digit. This control system can be modified to use any standard business machine cards with dimensional information.

Gaston E. Marbaix Ltd., of Devonshire House, Vicarage Crescent, London S.W.11 are the agents for the automatic bearingizing machine and the Barnesdril machine, and Automation Ltd., of the same address, for the Hautau automatic wheel rim machine.



Machining Aluminium Pistons

High-rate Production on the Tarex Automatic Turret Lathe

NOT infrequently elaborate and versatile machine tools are used mainly for relatively straightforward machining operations and the potentialities of the machine are far from being fully realized. For example, in the case of the Tarex single-spindle, automatic turret lathe,* it has been found that too often the user's planning or methods department approach a specific problem with what may be described as a conventional "screw-machine" attitude. Actually, the machine finds its best uses when considered as a precision turret lathe working automatically. When user and machine manufacturer collaborate in the investigation of a machining problem, quite outstanding progress may be made. An instance of the fruitful results of such co-operation has been the complete external machining of a die-cast aluminium-alloy piston in a single operation for a Continental automobile manufacturer.

The machine is the TAR-H, arranged exclusively for chuck work and equipped with an eight-station turret and a spindle-positioning device. Loading of the blank is done by hand, as also is the unloading of the finish-turned piston, but even these operations are facilitated by the machine. The accurate location of the blank and its clamping prior to the commencement of machining are effected as part of the machine automatic cycle.

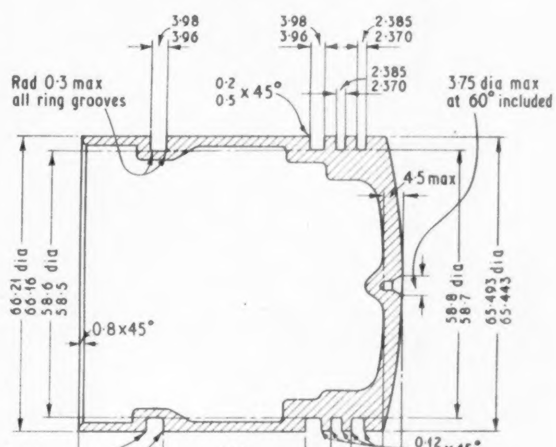
Clamping is performed by an air-operated, double-acting, expanding mandrel having two independent sets of jaws. Sequencing and timing of the work-gripping programme is controlled by the machine camshaft, with micro switches actuating electromagnetic valves. The blanks are cast to very close tolerances and internally the castings present a very intricate profile. These characteristics tend to facilitate the machining as only a relatively small amount of material has to be removed. On the other hand, they impose very difficult conditions for clamping since the run-out must be held to a very low permissible maximum. Any failure to maintain the work within this specified maximum run-out will result in uneven wall thickness and the production of scrap. Another difficulty encountered arises from the thin section of the piston walls. Excessive or ill-regulated clamping pressure can deform the castings and lead to the production of oval, tapered, or barrelled components that must subsequently be rejected.

Production was planned by the user's process engineers, bringing their experience of piston machining, in active collaboration with the Tarex tooling engineers, using their knowledge of the machine and its capabilities. First, the conditions under which the machine is operated may be noted. The spindle rotates at 2,551 rev/min, giving a maximum cutting speed of 550 m/min. In each minute of working time the spindle is braked from 2,551 rev/min to a standstill, remains stationary for a few seconds, and is then rapidly accelerated to the working speed of 2,551 rev/min. Obviously, only a robustly constructed machine can perform satisfactorily under such conditions. The spindle must be stiffly proportioned and well supported, and the carbide cutting tools must be rigidly mounted. The machining tolerances to be held on the work are given on the detail drawing of the piston. All dimensions are in millimetres.

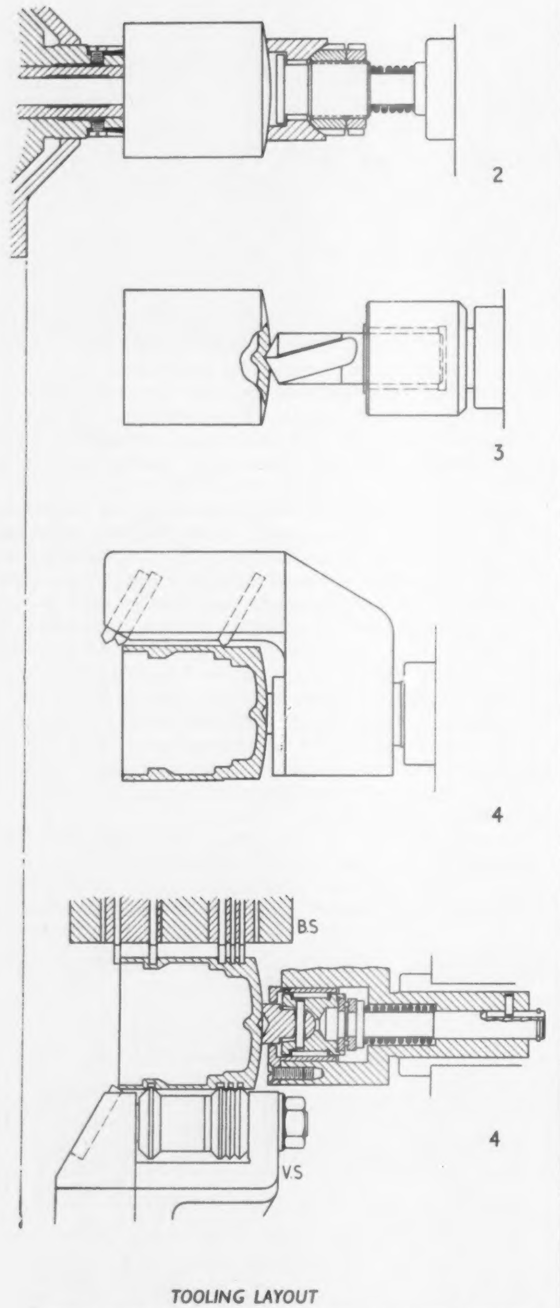
Specially developed for this component, the expanding mandrel has two air-operated, independently controlled sets of jaws. The rear set, that is the set closer to the machine spindle, comprises four evenly pitched jaws that can exert either a high-pressure or a low-pressure grip in the piston skirt. At low pressure, since they operate concentrically, they act more as a guide or as a steady than as a clamping means. Only two jaws, arranged diametrically opposite to each other in a plane at 45 deg to the planes containing the jaws of the rear set, are provided in the front set. Because of the unsymmetrical shape of the piston interior, one jaw has a single grip surface and the other two spaced and angled grip surfaces to give an effective three-point clamping action.

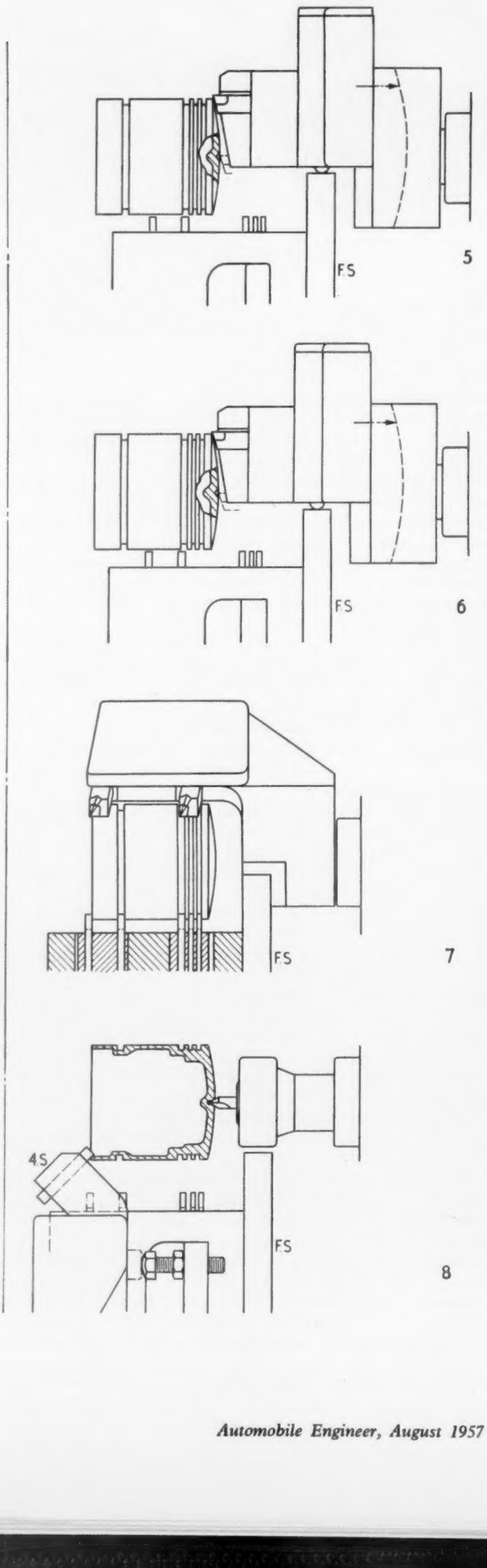
Flats are formed on each side of the mandrel to give clearance for the gudgeon pin bosses in the interior of the piston. This feature is used to advantage to locate the piston loosely in relation to the spindle and to position the

Machining details and tolerances of piston. All dimensions are in millimetres



*"Automobile Engineer." January 1957.

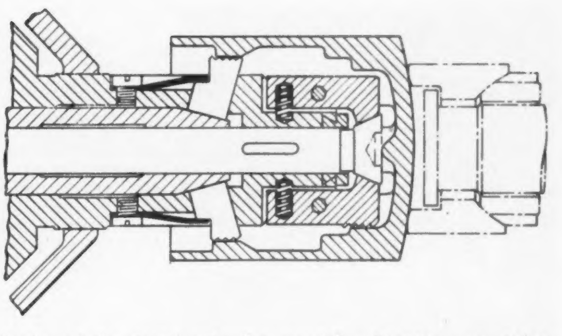




TOOLING LAYOUT

External machining of aluminium piston, 66·16 mm diameter. Spindle speed 2,551 rev/min. Cycle time 60 sec + manual loading and unloading

jaws opposite to their appropriate gripping surfaces in the piston. Since loading is performed by hand, the operation has to be regularized in order to reduce the time required to a minimum. Any fumbling or searching by the operator to ascertain the convenient angular position of the blank to enable it to be fed on to the mandrel would involve loss of time. The machine is equipped with the standard spindle-positioning mechanism and this is set to stop the spindle always in the same angular position. In this case, experience has shown that the most suitable position is that where the flats on the mandrel are vertical and the blank can be loaded with the gudgeon pin bosses lying horizontally.



Section of the special expanding mandrel, with piston mounted in position, showing rear and forward sets of jaws and the axial abutments

Schedule of Operations

Op.	Cutting speed m/min	Spindle speed rev/min	Stroke mm	Feed mm/rev
1	Hand loading			
2	Auto Chucking			
3	80	2551	3	0·1
4	550	,,	76	0·38
4 B.S	540	,,	5	0·05
4 V.S	,,	,,	(2	0·05)
5	,,	,,	34	0·13
6	,,	,,	34	0·13
7	,,	,,	52	0·2
7 F.S	,,	,,	(5	0·05)
8	35	,,	5	0·05
8 4 S		,,	(4	0·05)

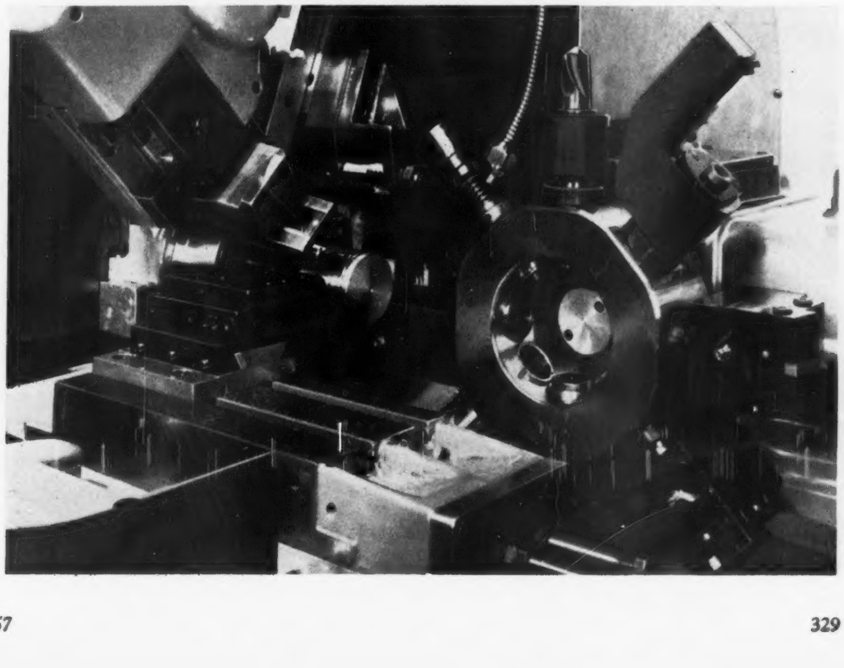
B.S—back slide; F.S—front slide; V.S—vertical slide; 4 S—fourth slide
Bracketed items relate to overlap movements that do not add to cycle time
Spindle rotation is left-hand, reverse of normal, throughout the cycle

From the front face of the mandrel project four tooth-shaped abutments, the function of which is to locate the blank axially. The form and arrangement of these abutments has been determined in order to suit the substantially flat surfaces remaining between the reinforcing ribs on the interior of the piston crown. To load, the operator feeds the blank on to the mandrel up to the four frontal abutments and pushes a button to actuate the front pair of jaws and clamp the blank. He then clutches-in the machine and the automatic cycle commences, first locating the blank to ensure minimum run-out and then completely machining it. The cycle time is 60 seconds, and during that time the operator can be occupied on any other task. Loading and unloading operations together take 10 seconds.

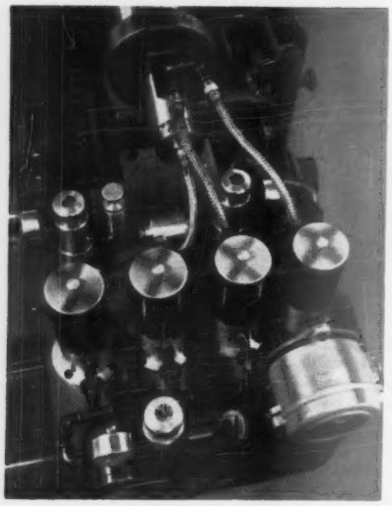
Automatic location of the piston blank on the mandrel is effected as follows. As soon as the machine is clutched in, the turret indexes from the free station, No. 1, to No. 2 station holding a hollow, hemispherically seated, spring-loaded stop and advances towards the work. Immediately the stop encounters the blank, the work spindle comes into motion and the front clamping jaws of the mandrel are relieved of their pressure. The piston blank is thus held only by its crown between the mandrel abutments and the hollow turret stop. Thereafter, the piston is firmly held by the automatic action of the mandrel jaws. First, the rear set of jaws grips at high pressure and then the front set, which is provided with only a single operating pressure, grips the piston close to the crown. The turret retracts and the work is then ready for machining. Operations are sequenced as follows:—

Operation 3. The casting burr at the top of the piston crown is removed and the centre is roughed by a large diameter drill.

Operation 4. The external diameters of the piston are roughed by radially-mounted tools. At the end of the turret stroke, a spring-loaded rotating steady mounted in the tool



Tooling set-up for completely machining the exterior of piston mounted on the special mandrel. Machine is shown at loading position, No. 1 station of the turret



Control equipment for the expanding mandrel, which has a high holding pressure for roughing operations and a lower pressure for finishing operations

holder engages the crown of the piston and consolidates the clamping of the casting on the mandrel. The steady, supported on a ball thrust race, runs in needle roller bearings and is furnished with a pin-retained, hemispherically-seated pressure pad. The back slide with blade-type tools then machines the four ring grooves to depth and roughs them for width. At the same time, a multiple circular tool on the vertical slide chamfers the external corners of all the grooves.

Operation 5. The crown of the piston is rough turned by a tool mounted in a special compound, profiling tool holder. The compound slide is cross-traversed by means of an abutment on the front slide.

At the conclusion of this operation, all roughing has been completed and only finishing operations remain to be done. Consequently, the component does not need to be held so firmly as previously. Accordingly, the pressure on the rear set of mandrel jaws is reduced to the lower value in order to lessen the risk of deforming the piston skirt.

Operation 6. Finish turning of the piston crown is done by a similar tool holder to that used in Operation 5.

Operation 7. The external diameters of the piston are finish machined with tangentially-mounted tools. At the same time, the ring grooves are finished to width by blade tools mounted in the front slide.

Operation 8. The piston is simultaneously centred at both ends. The centre in the crown is finished by a drill in a turret holder, while the internal edge of the piston skirt is turned at 45 deg by a carbide-tipped tool guided by a special tool holder mounted on the fourth slide and actuated by the front slide. This accurate centering is required in preparation for a subsequent grinding operation.

On completion of the machining operations, the spindle is rapidly brought to rest and positioned, as described earlier, to facilitate the unloading of the finish-turned piston and the loading of the next blank.

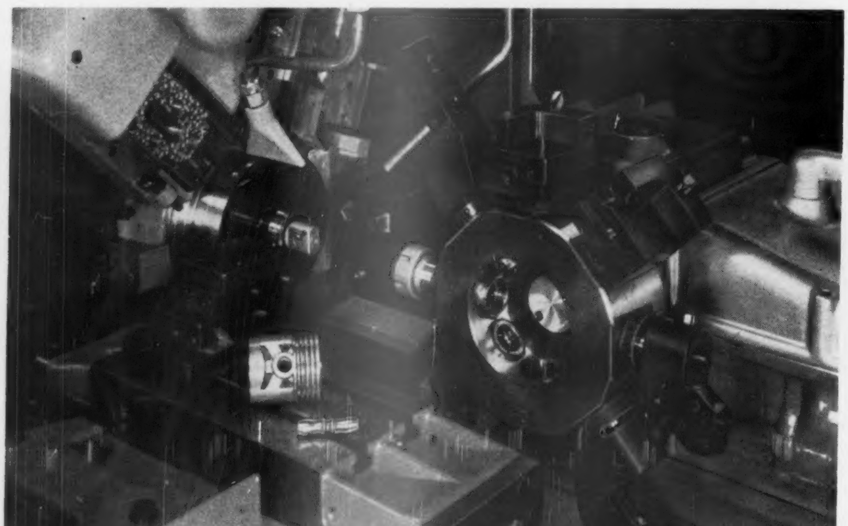
This tooling is performing to full satisfaction and has effected a substantial reduction in production costs. A single automatic turret lathe with one operator has replaced several simpler lathes, each with an operator. Floor space has been saved, a higher rate of production has been obtained, work is of higher quality and scrap has been virtually eliminated.

Due to the internal ribbing or configuration of pistons, it is not always possible to hold the component by means of the special expanding mandrel. Where necessary, use is made of a so-called "traction pin" and a drawbolt retractable through the spindle, as is illustrated. The piston skirt is registered on the short arbor and the pin is threaded through the gudgeon pin bores and the eye of the drawbolt, which is then drawn back to seat the piston on a shoulder on the arbor. As in the previous case, the spindle positioning mechanism ensures that the drawbolt is always presented in the same angular position to facilitate loading of the piston with the gudgeon pin bosses lying horizontally.

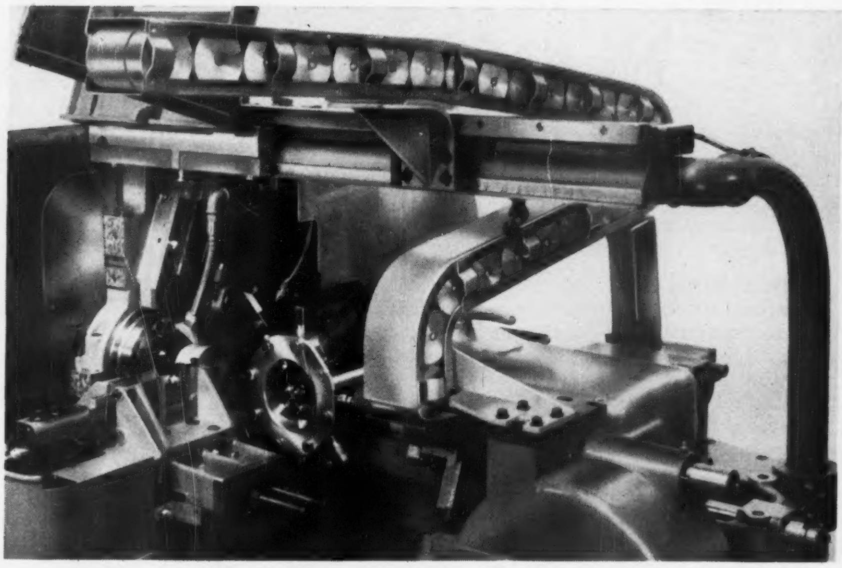
The sequence of operations on this piston is as follows:

1. Rough centre.
2. Finish centre.
3. Rough turn three external diameters.
4. Rough turn crown. Compound tool holder actuated by rear slide.
5. Finish crown. Compound tool holder actuated by rear slide.
6. Finish turn one external diameter.
7. Finish turn two external diameters.

The ring grooves are machined to depth and to rough width by tools on the rear slide. Chamfering of the ring groove



In this set-up, the piston is registered on the arbor and pulled back to a shoulder by a spindle drawbolt engaged by a traction pin through the gudgeon pin bores



Set-up for machining inside diameter of piston skirt. Loading, chucking, and unloading are effected automatically. Turret tools are duplicated and two pistons are machined in each machine cycle

corners is by a circular tool on the vertical slide and the grooves are finished to width by front-slide tools. The cycle time is 57 seconds, plus manual loading and unloading.

For certain machining operations it is possible to arrange automatic loading and unloading, thus enabling the machine to be run continuously and production rates to be further improved. An example of this practice is illustrated. Pistons requiring to be machined on the internal diameter of the skirt are charged in an inclined channelled track which loops from front to rear of the machine over the main turret slide and returns to the front of the slide. The components roll down the track by gravity and are delivered automatically to a loading station located in alignment with the machine spindle. From here a piston is picked up by a special split-sleeve type clip on the turret and, after four indexing movements of the eight-station turret, is transferred to the chuck which automatically clamps it for machining.

Machining is completed in two indexing movements and, after the next movement, another clip picks the component from the chuck. Two movements later, when the machined piston is depending vertically below the turret, a stationary cam on the turret axis is engaged by a plunger in the clip shank. This opens the clip and releases the piston which is deposited, skirt upwards, on the unloading chute or conveyor. The turret is equipped with a duplicate set of clips and tools, and while one piston is being transferred from loader to chuck, another is being machined. Obviously, with an output of two components per cycle very high production rates can be attained.

This Swiss-built automatic turret lathe is marketed and serviced in Britain by Tarex (England) Ltd., 22 Buckingham Gate, London, S.W.1.

Rubber Springs

According to M. A. Julien of Usines Paulstra, the crux of the problem of applying rubber springs in automobile suspension is the peculiar nature of rubber-type damping. Consideration of the case of forced vibrations in a mass/spring system with rubber-type damping and subjected to a disturbing force indicates that the excess pressure in a given suspension with rubber-type damping is independent of the sprung mass. For a given amplitude of the disturbing force, the maximum amplitude of resonance is constant whatever the sprung mass (and, consequently, the natural frequency of the system). It also varies proportionately to the amplitude of the force imposed. The logarithmic decrement of free vibrations and the damping and resilience of a suspension with rubber-type damping are constants depending on the spring and independent of the sprung mass.

Damping in vehicle suspension must control the amplitude of vibrations of relatively low frequency in the sprung masses by combating resonance on wavy-surfaced roads, and also restrain relatively high frequency vibrations of the unsprung masses to ensure uniform road holding. These two requirements are inseparable when entrusted to a single device working in parallel with the main suspension spring between the wheel, or the axle, and the sprung masses. They may be met by a viscous-type shock absorber of constant velocity. Damping of the rubber type, even though adequate for correct control of the vibration of the sprung masses is much less effective in respect of the unsprung masses.

With rubber-spring suspensions, even if the suspension itself may seem satisfactory, road-holding will be definitely unsatisfactory, especially at high speeds. It is agreed that the average optimum coefficient of damping should be at least one-quarter or one-fifth of the critical damping, and although new natural-rubber-based mixtures give more promise in this respect, the best of those now available can provide no more than one-third, or at most one-half, of this value. Rubber should, therefore, be used only with a supplementary damper capable of supplying between one-half and two-thirds of the damping of the actual suspension and an even greater proportion of that of the unsprung masses. A solid-friction shock absorber, alone or in conjunction with a simplified hydraulic damper, is suggested in this connection.

Resilient Bushes and Pads in Suspension Systems

The Use of Rubber for Vibration Isolation and to Reduce the Number of Lubrication Points

FOLLOWING the widespread adoption of independent suspension, rubber bushes have been increasingly employed to reduce the number of lubrication points. Another marked advantage of rubber bushes is that they eliminate the need for maintenance and replacement. Provided they are properly designed for the application and are correctly fitted, they do not wear as do metal bushes. There are examples of rubber bushes that have operated without attention for as long as 25 years. By virtue of its resilience, rubber also has the advantage that it tends to reduce the transmission of vibration to the vehicle structure—a most desirable feature where chassisless construction is employed. In fact, rubber components are, of course, sometimes used solely for the purpose of vibration isolation.

Rubber is not without its disadvantages for certain applications. Some manufacturers consider that, owing to its resilience, it is not suitable for pairs of bearings that are closely spaced in independent suspension systems, for example, the outer bearings of lower wishbone arms, particularly when the inner bearings are rubber-bushed. This is because the outer bearings are subject to brake torque and drag loads, which tend to cause deflections that might interfere with the steering geometry. Although the inner bearings are also subjected to these loads, their deflections are not so serious, because of their relatively wide spacing. Provided that the angular deflection required about the major axis of the bush is not too great, the amount of radial deflection can be regulated to a large extent by restriction of the thickness of rubber employed. Another reason why in some instances it is not practicable to employ rubber bushes for outer bearings of wishbones is that the space available may be restricted because of the need to provide clearance for the movement of the wheel to the extreme positions of its steering lock.

Although objections to rubber bushes, on the grounds that they squeak, have been made, it seems probable that they are not valid. In every instance when complaints of squeaking rubber bushes have been investigated by Silentbloc Ltd., the noise has been traced to components other than the rubber. The cause has generally been found to be interleaf friction of the rear springs, or loosening of the fixings of the bushes, resulting in relative movement between the metallic components.

Materials

Mechanical engineers in general find it difficult to understand the limitations of rubber as an engineering material. This is because its performance varies widely not only with temperature, but also with rate of stress application. Therefore, the choice of material for a specific duty is based on complex factors. In addition, tolerances on the dimensions of the rubber components in bushes, generally have to be greater than $\pm 0{\cdot}003$ in, and spring rates may vary by as much as ± 15 per cent. Closer tolerances can in many instances be maintained only by selection, and therefore may be uneconomic. This lack of precision is, perhaps understandable when it is remembered that the molecular chain-structure of rubber is much more complex than the crystalline structure of metals. However, it should be remembered that by comparison with the extensibility of the material, the tolerances are not so large. It is of interest that rubber is the only material that can be made in such a manner that it can be extended 1,000 per cent and recover almost immediately to its nominal dimensions.

In general, British manufacturers use natural rubber, while many of the American manufacturers employ synthetic rubber for bushes. There appears to be no real advantage of one material over the other, except perhaps price stability, which is in favour of the synthetic materials. A synthetic material with properties like those of natural rubber has recently been discovered, but is not yet manufactured in large quantities. It is of interest that over the last twenty years the price of natural rubber has fluctuated between about 7d and 72d per pound.

Although some synthetic rubber does not deteriorate readily when exposed to contamination by hydrocarbon oils, this is not an important factor so far as bushes are concerned, because in these components the area of rubber exposed to contamination is very small. A few isolated instances of bushes having deteriorated as the result of contamination by oil have occurred. However, it is suspected that the trouble was caused not by normal exposure but by the components having been inadvertently saturated over a long period in paraffin or oil. Needless to say, rubber bushes should not be soaked at all, but it is possible that inexperienced mechanics, not realizing that the bushes are of rubber, might put them in the paraffin bath, together with the other components of an assembly. In normal service, it has been found that the coating of mud that inevitably becomes deposited on rubber bushes used in suspension systems tends to protect the rubber against attack by oil or grease. In any case, even in applications where bushes are fairly heavily contaminated, it has always been found that the depth of penetration is relatively small. If necessary, a varnish, such as Vulcaprene or Hypalon, can be used to protect the surface from oil penetration.

A selection of flanged bushes suitable for use in suspension systems

Synthetic rubbers are made from oil, acetylene, coal,

limestone, ammonia and hydrochloric acid. They do not have such a high fatigue or permanent-set resistance as natural rubber. However, these disadvantages can sometimes be offset by suitable design measures. Other materials that have been tried in rubber bushes are butyl and silicones. Butyl has good damping characteristics, but is relatively poor so far as fatigue resistance is concerned and is liable to take a permanent set more readily than natural rubber. The silicones are outstandingly resistant to high and low temperatures, but are markedly inferior to the more conventional materials in almost every other respect.

Conditions of operation

So far as climatic conditions are concerned, there are three factors that affect the performance of rubber. These are temperature, light, and ozone concentration. At operating temperatures higher than those generally encountered in this country, the rate of deterioration may be accelerated. Evidence of this deterioration is reduction in tensile strength, tear and abrasion resistance, and elasticity. High temperatures, which may be developed as a result of hysteresis, have been known to melt rubber: for example, in military vehicles, solid tyres have become so hot internally that the rubber has vaporized and the tyre exploded, spattering the vehicle with the molten compound. In general, the rate of deterioration is greater if the high temperatures occur under conditions of high relative humidity. Provided the temperatures are not too great and the loading is light, the rubber may not show signs of failure, but after a period of perhaps twenty years it might become hard, dry and brittle.

Low temperatures increase the modulus of elasticity. This is understandable, since deformation is partly elastic and partly plastic. It is, of course, the plastic component of resistance that markedly increases as the temperature is lowered. At about −42 deg C, the stiffness of the material is generally almost doubled, and between about −50 and −55 deg C, the rate of increase of stiffness changes rapidly and the material tends to become leathery and almost brittle. When the temperature is less than −62 deg C, the rubber crystallizes and becomes positively brittle. Synthetic rubbers, except silicones, behave in a similar manner and have higher freezing points than natural rubber.

A little-known property of rubber is that if it is maintained at freezing point or slightly less, it tends to become stiff after long periods of exposure, measured in terms of years rather than months. This is because the material crystallizes—in its normal state, rubber has some properties resembling those of a supercooled liquid. The normal elastic properties of the crystallized rubber are restored when its temperature is raised above freezing point.

Light causes surface cracking, sometimes termed light-crazing. White rubber is particularly susceptible to this defect. However, the effect of light on rubber is not important, so far as bushes are concerned, because such a small area of the material is exposed.

Ozone is only seriously detrimental to rubber when the material is stressed. In fact, its effect is at a maximum when the surface deformation is between about 15 and 25 per cent. It follows, therefore, that deterioration is not directly proportional to stress; however, for a given level, it is directly proportional to ozone concentration. The combined effect of ozone and stress on rubber is the development of surface cracks at right angles to the direction of the tensile stress. Fortunately, when cracking occurs, the stress is relieved locally so that there is a tendency for the damage to be restricted. So far, little data has been accumulated on local ozone concentrations. Therefore, at present there is no point in developing special ozone-resistant rubbers for any particular parts of the world.

Some synthetic rubbers are more susceptible than the natural ones to ozone cracking. It is not surprising, therefore, that in the United States of America, where synthetics are widely used, additives have been developed which, when dispersed in the rubber, prevent ozone cracking. A disadvantage of these additives is that they tend to migrate and stain paintwork. For this reason, they cannot be used, for example, in glazing strips on cars. In any case, they are relatively expensive and are not widely employed, except in the more susceptible, general-purpose synthetics used in the United States. The coating of mud that tends to cover rubber in suspension components gives some measure of protection against the ill effects of ozone.

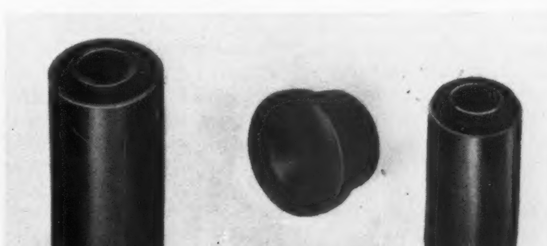
Practical implications

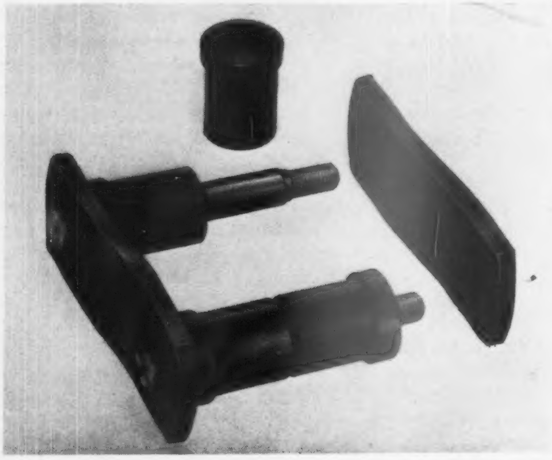
To obtain satisfactory results from rubber components it is, of course, necessary to take into account the properties already discussed. For example, if rapid, large amplitude movements are to be expected, the rate of generation of heat may be high. Therefore it is necessary to design the component in such a way that the heat can flow readily away from the rubber. Since rubber is a bad conductor of heat, its thickness should not be too great and its surface area in contact with metal should be as large as practicable.

Fortunately, heat generation in bushes used in vehicle suspension systems is not a serious problem in this country. This is not only because ambient temperatures generally are moderate, but also because road surfaces are good and therefore excessive deflections are not experienced continuously for long periods. Even in countries where pavé roads are common, the problem of heat dissipation hitherto has not been so great as might be expected, since drivers tend to restrict their speeds to avoid undue discomfort that would be caused by excessive deflections of the suspension system. However, the situation might have to be reviewed in the near future, because new suspension systems are being developed to enable vehicles to travel faster over rough terrain without causing discomfort to the occupants.

It might be asked why rubber bushes can be employed in very cold countries. Probably the answer to this is that hysteresis increases as the temperature is lowered and so the rate of heat generation also increases. Therefore, the temperature of the bush tends to rise rapidly to a level at which its resilient properties are such that it can operate normally. In general, rubber components can be operated satisfactorily in temperatures of between −40 and +60 deg C.

Left: A plain Silentbloc bush. Centre: The flanged taper bush with a flanged inner sleeve. Right: Extended-rubber bush, designed to give good recovery from axial loading. Below: A plain taper bush, which is compressed axially on assembly to form a flange at one end





Silentbloc Harris C.P. shackle assembly. The shackle plates are domed to compress the flanges in such a way that they do not tend to spread

In most applications, the bushes are arranged so that their outer sleeves are a force-fit in their housing and their inner sleeves are clamped securely by the central bolt, or pivot pin. Where the loading is particularly heavy, such as in commercial vehicles, the inner sleeve is usually positively fixed. It may have a tongue to register in a slot in the bracket or it may be keyed to the pin. Alternatively, the central component can be a solid pin instead of a sleeve, and its ends are securely fixed. For more lightly stressed applications, bushes without a metal outer sleeve are sometimes employed; the rubber is compressed between the inner sleeve and the bore of the housing. Bushes should be assembled and clamped with the suspension in the normal position, so that equal movement is obtained in both directions.

So far as loading is concerned, it is not possible to lay down hard and fast rules, but the following can be used for general guidance. It is desirable to restrict the journal loading in the static condition to a value of from 300 to 400 lb/in^2, based on the projected area of the inner sleeve. However, in many applications, bushes are operated satisfactorily at static loadings of about 500 to 600 lb/in^2. Dynamic loading may be from seven to ten times these values, but rubber appears able to withstand shock loads remarkably well. Possibly this is because a certain length of time is required for the plastic deflection to take place and, under shock loading conditions, the load is not applied long enough for the maximum deflection to be realized, so that the strain is not great enough to damage the material.

An important feature, so far as radial load-carrying capacities are concerned, is the ratio of the area of rubber exposed at the ends of the bush to the bearing area. If this ratio is high, the rubber can be relatively easily squeezed out from between the inner and outer sleeves and its load-carrying capacity, therefore, is not so great. The tendency for the rubber to be squeezed out can be reduced by turning the ends of the outer sleeve inwards, as in the illustration of a bush designed for the accommodation of heavy axial loads experienced on public service vehicle suspension systems. Alternatively, the shackle plates can be clamped against the ends of the rubber to compress it into the bush. If these shackle plates are dished locally, as in the accompanying illustration, they are even more effective in retaining the rubber in the bush.

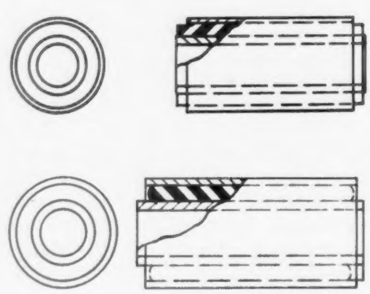
In almost all applications, the rubber is precompressed between the inner and outer sleeves, because this increases its capacity for carrying torsional loads. However, in some instances bushes, in which the rubber is bonded to the inner and outer metal sleeves with no precompression, are used. The thickness of the rubber used in the bush is determined by the angular deflection required and the amount of radial deflection that is permissible. Radial deflection can be regulated to a certain extent by the hardness of the rubber mix. Regulation of the precompression of thick rubber bushes is limited by practical considerations of the volume of rubber to be compressed.

Because of the part plastic and part elastic deformation of rubber, its dynamic modulus, that is, load deflection characteristic, is different from its static modulus. However, this difference is only important when the load is applied rapidly. The rubber mix can be varied according to requirements, for example, some grades are particularly suitable for vibration damping because their hysteresis reduces the amplitude at resonance. Butyl has a good damping characteristic, but is a dead rather than an elastic material and is not a good stress carrier. It is useful, however, for applications where resonance cannot be avoided, for example, in certain machines that have to be run through the resonance period when they are started up.

Silentbloc Ltd. state that experience has enabled them to develop a range of bushes, each of which is designed to suit a certain type of application. This is convenient for vehicle designers, because they can generally find a standard production bush to suit their requirements. However, it is essential to study each new application to ensure that there is nothing extraordinary about its requirements that would call for special design features in the bush.

Applications

A number of different types of bush and some other components are shown in the accompanying illustrations. One is the original Silentbloc bush, which comprises an inner



An extended-rubber bush is shown above, and below is the plain Silentbloc bush

and outer sleeve with the rubber precompressed between them. The precompression is effected by stretching the rubber cylinder during its assembly into the two metal sleeves. When the stretching force is released, the rubber endeavours to regain its free form, and this exerts radial pressure on both the inner and outer sleeves. The inner sleeve is longer than either the rubber or the outer sleeve and is clamped by the pin. In most applications, the outer sleeve is pressed into its housing.

A development from this component is the extended-rubber bush. The rubber extends beyond the ends of the outer sleeve and is terminated just short of the ends of the inner sleeve. When this bush is assembled into position, the end plates, or shackles, are tightened against the inner sleeve by the central bolt. The advantage of this arrangement is that, under axial loads, the rubber ends are compressed to form small flanges, so that when the load is released recovery is greatly assisted.

The next step was to develop a flanged bush to provide more positive and controlled reaction to axial loads. This

type of bush has one end of its outer sleeve and the rubber component flanged, but its inner sleeve is not flanged. So far as radial loading is concerned, the rubber is, of course, precompressed between the inner and outer sleeves. It is important to regulate the thickness of the rubber flanges according to the angular deflection required, otherwis they may be overstressed in operation. Like the other bushes of this type, two are generally used together, assembled one in each end of a common housing. The precompression in the flange is regulated by the difference between the axial dimensions of the housing and the inner sleeve.

Flanged bushes can be provided at a lower cost by omitting the outer sleeve and tapering the rubber. This type of bush is used in a tapered housing, the rubber being squeezed into the taper by the central retaining bolt, to apply precompression. The degree of precompression is important if slip, on the one hand, and overloading, on the other, are to be avoided. This type of bush is commonly employed in independent front suspension systems.

Some rubber bushes have only a plain cylindrical inner metal sleeve and a flanged rubber component. The rubber components are of tapered form, and the bushes are assembled one into each end of a double-tapered housing. The degree of axial precompression is regulated by the difference between the lengths of the inner sleeves and the housing, while the radial precompression is determined by the taper and the axial movement of each bush into the taper. In the accompanying illustration, of this type of bush, it can be seen that the inner face of the flange is not at right angles to the axis of the bush but is chamfered to form a fillet between the periphery of the flange and the periphery of the main portion of the bush. This detail feature is incorporated to ensure that tension stresses are not developed locally between the flange and the main portion of the bush. It is desirable, in order to prevent wear and deterioration in this area.

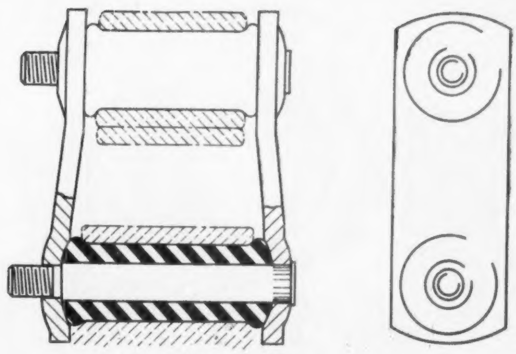
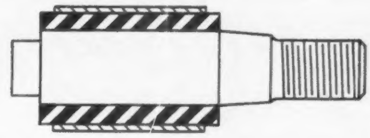


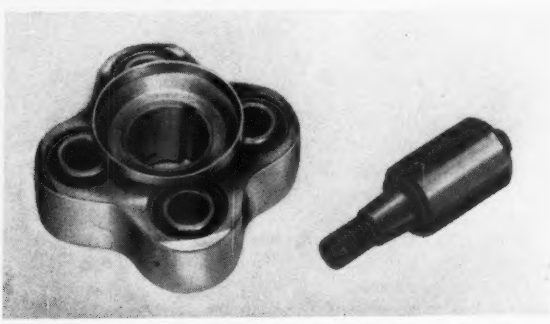
Illustration showing a section and elevation of the C.P. shackle



Rubber bushed fittings of this type can be used on track rod ends

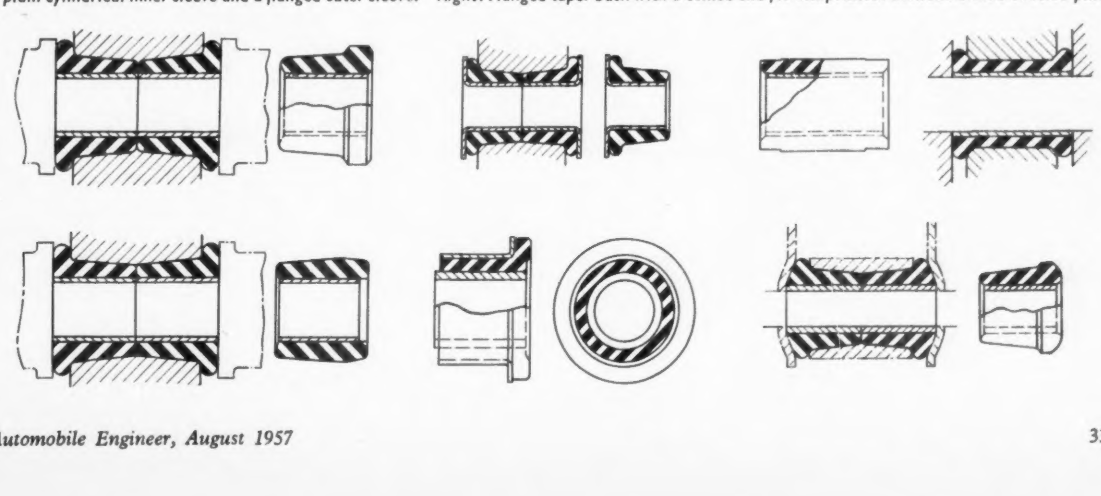
Two extremely low-priced bushes have been developed from those already described. One is the plain taper bush. This comprises simply an inner sleeve surrounded by a tapered rubber bush. One end of this bush projects beyond the end of the metal inner sleeve. These components, like the others, are used in pairs, with their projecting rubber ends outermost. When the central bolt is tightened, the tapered bushes are forced into their tapered housings and the projecting ends of the rubber bushes are compressed in such a way as to form flanges to take axial loading. Since the flanges are not preformed, a larger degree of axial compression is required than with the other types: therefore, more space is required for its initial assembly.

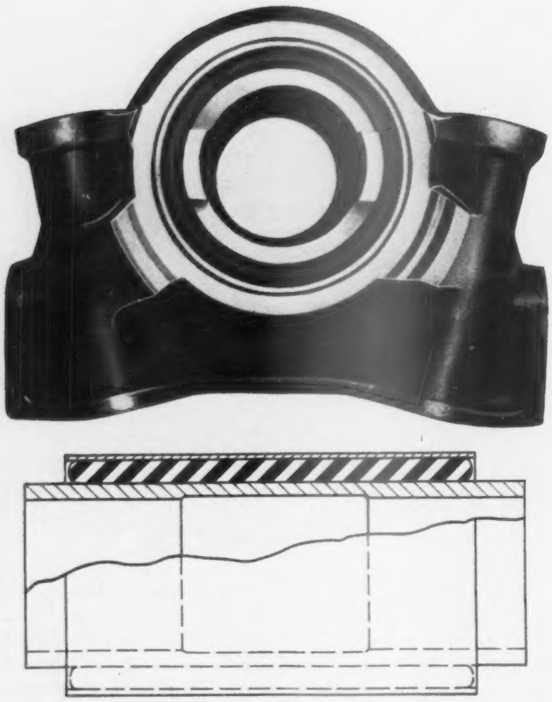
The second low-priced bush is a plain cylindrical component, termed by Silentbloc a N.O.S. bush. It has an inner metal sleeve surrounded by a slightly longer rubber bush. At assembly, a special funnel-shaped tool is used, through which this bush is forced into its housing in the chassis component. The main portion of this housing is of plain cylindrical form, but its ends are chamfered to allow for the formation of a flange when the ends of the rubber component are compressed by the tightening of the central bolt. The advantages of this design are that, having no outer sleeve, it is inexpensive and its space requirements are less than those of a bush with both inner and outer sleeves. However, it has



Above: A flexible coupling used between the steering column and the steering box of a military vehicle, and a rubber bushed, track rod end-fitting

Below: Selection of rubber bushes. Top, left: Flanged taper type. Centre: Flanged taper bush with flanged inner sleeve. Right: Silentbloc N.O.S. bush. Bottom, left: The plain taper bush, which is compressed axially in its housing to form the flange. Centre: A flanged Silentbloc bush, with a plain cylindrical inner sleeve and a flanged outer sleeve. Right: Flanged taper bush with a domed end for compression between dished shackle plates





Above: Two illustrations showing a rubber bush type, trunnion bearing for the suspension bogie of a heavy commercial vehicle

the disadvantage that the amount of radial precompression that can be applied is less than that which is possible with other types of bush.

A disadvantage of flat-ended, flanged bushes is that when the flanges are compressed, the rubber tends to be squeezed radially outwards, so that tension stresses may be formed locally round the periphery of the flange; also, the degree of precompression that can be applied to the main portion of the bush is limited. To overcome this difficulty, dome-ended, rubber flanged, tapered bushes have been developed. These bushes are used in pairs, as are the other tapered bushes. Not only are their ends domed, but the shackle plates between which they are compressed are cupped, so that when the central bolt is tightened, the rubber tends to be forced into the housing and a relatively large degree of precompression can be effected.

For commercial vehicles, where loading is particularly heavy, special bushes have been developed. One of these is shown in an accompanying illustration. It comprises a central pin and an outer sleeve with inwardly-lipped ends, the rubber being between them. The central portion of the pin is shaped so that it is approximately parallel to the lipped outer sleeve, and its ends are increased in diameter and securely clamped in their housings. The reason why the ends of the outer sleeve are lipped inwards is to enable the bush to take axial loading and also to help retain the rubber in the component when heavy radial loads are applied.

The other illustration of a rubber bush for commercial vehicles shows the 11 $\frac{9}{16}$ in long trunnion bearing at the centre of a spring for the Leyland Hippo four-wheel bogie. An advantage of this type of bearing, as compared with the roller or needle roller type bearings commonly used, is that it accommodates rocking in transverse planes. This rocking tends to impose excessive edge loads on rolling-element type bearings; moreover, difficulties may be experienced with rolling element bearings because of the limited angle of oscillation in this application. Although the float which this type of bearing allows is just sufficient to accommodate the rocking, the flexibility of the bush is limited so that steering and stability are not adversely affected.

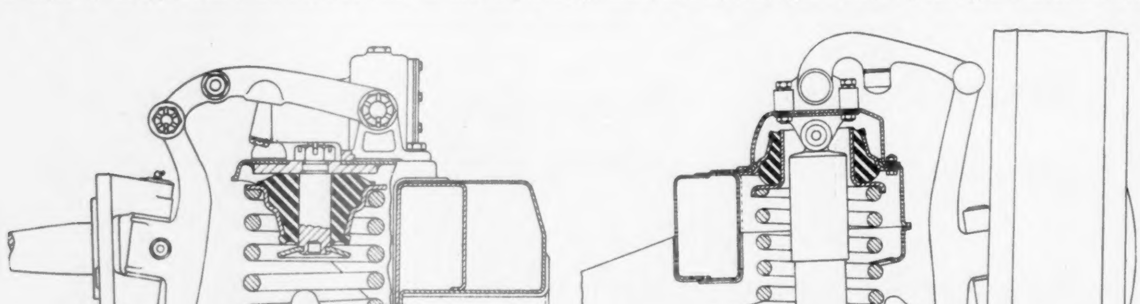
In the accompanying line drawing of the trunnion bearing, the ends of the inner sleeve are shown to be of plain cylindrical form. However, it is usual for manufacturers to machine some of the metal from these ends, leaving a pair of tongues on each, which register in slots in the end fittings of the trunnion. This is desirable with heavily loaded bushes, because the clamping action alone of a nut on the end of the inner sleeve is not sufficient to prevent relative movement between the sleeve and the trunnion pin.

Other applications for rubber in suspension and steering

Since noise may be transmitted from the lower wishbone arm of a suspension unit, through the coil spring to the vehicle structure, it is desirable to interpose rubber between the spring and the structure. One way of doing this is to employ a unit such as the Silentbloc Frustacon mounting, illustrated here in two forms. One is for use in systems in which a telescopic shock absorber is carried coaxially with the spring, while the other is for the alternative arrangements and is illustrated with a bump stop mounted coaxially with the spring.

Both units comprise three components: an outer steel pressing, with a circular flange round it to replace the seat for the coil spring; a steel inner sleeve, or a bolt; and a conical rubber insert, separating the two steel components. As can be seen from the illustrations, the rubber is so shaped that

Examples of the application of the Silentbloc Frustacon mounting to independent front suspension systems with different types of shock absorber



light loads are taken mainly in shear, but as the loading increases, a greater proportion of it is taken in compression. The unit can be fitted either above or below the spring, according to which is the more convenient arrangement.

In neither system are the space requirements excessive. When a telescopic shock absorber is coaxial with a spring, it generally governs the overall height of the suspension system, so the Frustacon unit can be mounted entirely above or below the spring, without taking up much more space. However, with the other arrangement, the alternative unit is generally mounted in such a way that most of its bulk is inside the spring, so that little space is wasted.

Bump stops, of course, are generally of ovoid shape, to give a rising rate as their deflection increases. They are usually designed in such a manner that they are relatively lightly stressed, to restrict the strain to about 50 per cent. Expressed in terms of the area of the base section, the maximum stress is generally in the order of about 200 lb/in^2.

Another method of preventing the transmission of road noise through front suspension systems to the vehicle body is to mount the suspension and steering assembly on a substantial cross member and to interpose rubber between this cross member and the main structure. The arrangement frequently adopted is to have two conical mountings, with their axes vertical, one on each side above a transverse line joining the wheel centres, together with one or more rubber bushes in a more forward position and with their axes horizontal in a transverse plane. With this arrangement, the conical mountings absorb high frequency vertical vibrations, while the horizontal bush, or bushes, restrict fore and aft movement.

In some vehicles, rubber pads are interposed between the leaf type rear springs and their steel seatings. These pads are generally reinforced with fabric and one small passenger car, currently in production, has a fabric-reinforced pad of this type, which is stressed under static conditions to 625 lb/in^2.

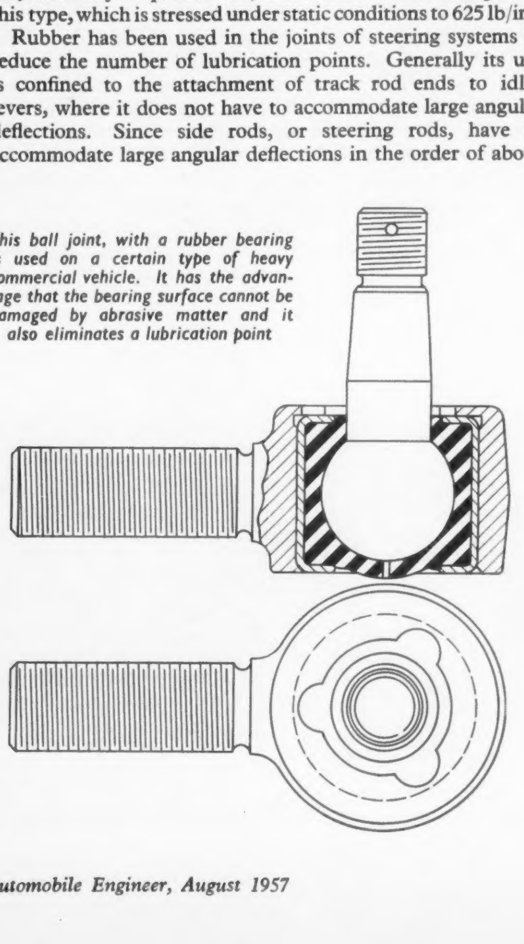
Rubber has been used in the joints of steering systems to reduce the number of lubrication points. Generally its use is confined to the attachment of track rod ends to idler levers, where it does not have to accommodate large angular deflections. Since side rods, or steering rods, have to accommodate large angular deflections in the order of about ± 35 deg to allow for steering movement from lock-to-lock and vertical deflection of the suspension, conventional ball joints are almost universally favoured for their end fittings. However, a steering ball joint has been operated successfully, as a replacement for a conventional ball joint, on a certain type of commercial vehicle. This ball joint is shown in an accompanying illustration. Its principal advantages are that lubrication is eliminated and, because the rubber is pre-compressed, there is no danger of abrasive dust getting into it and causing wear.

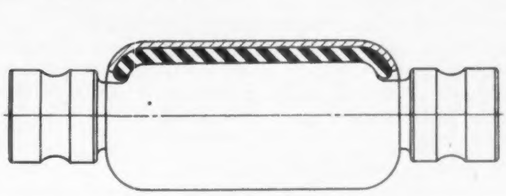
On assembly, the cup-shaped rubber is placed in a pierced, cupped housing and the ball component pressed in. Then the upper edges of the pierced cupped housing are turned in to retain the rubber in its precompressed state. The small hole in the base of the rubber cup is, of course, to allow air to escape when the ball is forced in. This assembly is then pressed into the eye of the end fitting of the rod.

An accompanying illustration shows a coupling that is fitted between the steering column and the steering box of a military vehicle. However, this type of coupling would generally be regarded as too expensive for commercial vehicles or private cars. The main requirement for couplings in this type of application is that, should the flexible element fail, there is still a direct mechanical drive available. In the unit illustrated, even in the unlikely event of the rubber bushes failing, the steering connection is still maintained between their inner and outer sleeves.

Another application for rubber components is the universal unit at the upper end of the suspension strut of the Ford Zephyr and Consul models. The function of this unit is to provide for universal action of the suspension strut under steering and riding conditions. It is also designed to insulate road noises from the body. An advantage of this device is that the universal action is obtained without need for lubrication; it also facilitates assembly. From the illustration, it can be seen that the lower end of the rubber is lipped to seal against the spring pan.

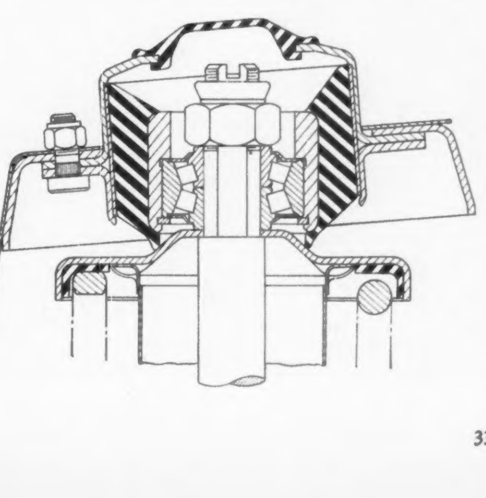
This ball joint, with a rubber bearing is used on a certain type of heavy commercial vehicle. It has the advantage that the bearing surface cannot be damaged by abrasive matter and it also eliminates a lubrication point





This bush for heavy commercial vehicles has a rubber sleeve between an inwardly-lipped outer sleeve and a barrel-shaped pin

The rubber unit installed at the upper end of the front suspension strut of the Ford Consul and Zephyr models is designed to provide for universal action under steering and ride conditions



TORCH CUTTING AND WELDING

Improved Equipment for Use on Aluminium and Stainless Steel

EXPERIENCE has shown that torch equipment which is completely satisfactory for ordinary steels is not altogether suitable for light alloys and stainless steels. Because of this, British Oxygen Gases Ltd., Spencer House, St. James's Place, London, S.W.1, have recently introduced three new items of equipment; one for cutting and profiling aluminium, one for Argonarc spot welding and an arc-length control unit. Mention may also be made of this Company's development of flame cleaning apparatus, although this is a matter of interest to the plant engineer rather than the production engineer.

Argonarc spot welding

The Argonarc spot welding process, which was introduced several years ago, made it possible to effect a joint between certain light gauge materials from one side only. A disadvantage of the original process was that it proved inconsistent in operation and resulted in cratering and cracking of the fused zone. To prevent this, the weld must be cooled slowly, and British Oxygen Gases Ltd., have now introduced a system of control that effectively eliminates cratering and gives consistent results.

A D.C. rectifier is used as a power source, and crater control is effected by the introduction of resistances into the magnetic amplifier control circuit. This crater device reduces the current to zero in four stages. In addition to the D.C. power source, the equipment used includes a timing control and a torch. Normally the torch is held in the hand, and is used to position the electrode above the material and supply the argon that shields the weld. A trigger switch incorporated in the gun handle sets the timing circuit in motion; the arc can then be struck for a predetermined period. The heat from the arc flows over the inter-face and the fusion of the materials is completed. A sound joint, with a high strength value, is obtained without the use of filler metals.

A complete development programme has been carried out on the 18/8 group of stainless steels with a view to getting the maximum possible strength and a high degree of consistency with all spot welds. Strengths of the following order can be expected on 18/8 stainless steel:—

Material gauge	Shear strength, lb per spot
24	450
22	690
20	1100
18	1300
16	1600

The process is also applicable to some new high tensile steels, including FV520 and FSM1.

Automatic arc-length control unit

A constant length of arc is very desirable in high quality welding, and for this purpose British Oxygen Gases Ltd., have produced the unit, shown in the accompanying illustration. This unit includes the new Mark IV 600 amp, water-cooled torch with back-loading collets and threadless nozzles. The head is robust and gives a steady torch position whether filler wire is used or not. A remote control panel, including a reference voltmeter, is provided so that the equipment can be operated from the welding position by remote switches. The main control panel has facilities for A.C. or D.C. welding and the unit can be used with an alternative head for consumable electrode welding.

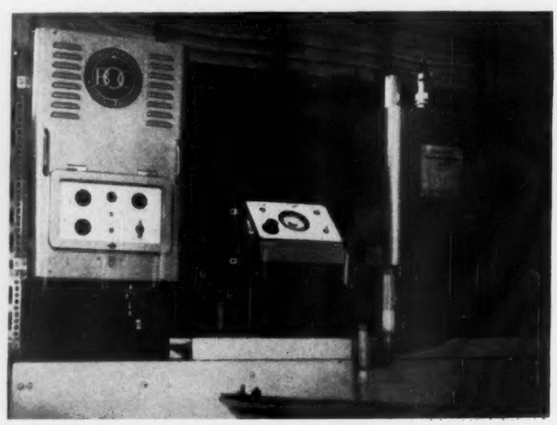
This equipment can be used for longitudinal and circumferential welds in light gauge material which must be jigged and held accurately to make an efficient weld. Once the arc length is preset, it is automatically maintained. This will be of particular value in welding components of varying contour.

Cutting aluminium

The Argonarc cutting process has been introduced for cutting and profiling aluminium. It enables aluminium plates up to 1 in. thick to be cut, using either oxygen-free nitrogen or other gases. The arc cutting action is produced by a constricted arc of a high temperature and velocity being struck between a tungsten electrode and the surface of the metal and surrounded by a shielding gas. The concentrated and columnated energy of the arc stream melts and ejects a thin section of metal to form a kerf. A clean cut edge, with little or no adhering dross, is obtained. A power source giving an open circuit voltage of at least 110 volts enables arc voltages of 66-80 to be obtained at various current settings.

Argonarc spot welding equipment

Automatic arc-length control unit



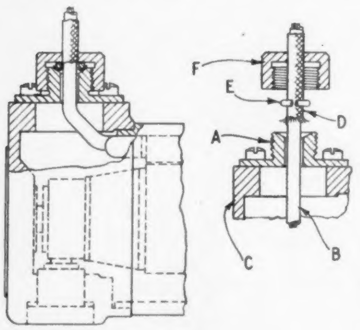
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Securing electrical leads

The problem of establishing the secure connection of a cable to an electrical component so that the conductors are relieved of strain is of long standing. The aim of the invention is to achieve such a connection with the incidental advantage of grounding the metallic sheathing of the cable. It is claimed to meet the requirements of mass production methods of manufacture and assembly and to be able to withstand such abuses as, say, the carrying of an electric motor by means of the cable.

A screwed and coned bushing A, apertured to pass the cable B, is flange-mounted on the motor casing C. The metallic



No. 758978

braiding or sheathing D of the cable is trimmed and unravelled for an appropriate distance in order to overlay the coned seating of bushing A. Threaded over the cable and brought down into contact with the sheath ravellings in the coned seating is a split ring E. An apertured cap nut F draws down upon the ring and clamps the ravellings, thus securing the connection. The conductor is relieved of all stress in handling or assembly and the metallic sheathing is effectively grounded on the motor casing. *Patent No.* 758978. *Borg-Warner Corp.* (*U.S.A.*).

Coated piston rings

Cast irons of special compositions are commonly used in the production of piston rings. In the selection of materials the aim is to secure optimum values of elasticity, resistance to wear, and resistance to corrosion in the same composition. According to the invention at least those surfaces of a ring subject to sliding friction are provided with a coating of a nitride, boride or silicide of titanium applied by reaction from the gaseous phase.

Precipitation of titanium nitride is usually effected at temperatures over 1,000 deg C, but it was discovered that temperatures in excess of 1,000 deg C are not necessary for the application of titanium nitride coatings. Despite the diffusionless growth on the parent metal, coatings in thicknesses between 1μ and 100μ can follow deformations of the parent metal without peeling off.

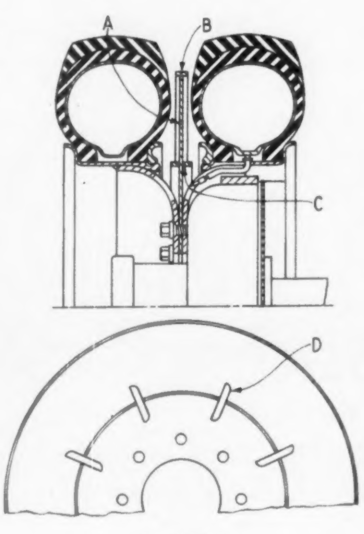
Other advantageous properties of the coatings are good adhesion of lubricating oil; no tendency to seize under unfavourable conditions of, or absence of, lubrication; good running-in characteristics; and only slight wear on the co-acting metal. Similar behaviour is shown by coatings made from titanium boride and titanium silicide which may also be precipitated by reaction from the gaseous phase. *Patent No.* 760664. *Metallgesellschaft A.G.* (*Germany*).

Protecting tyres on twin wheels

Pieces of stone, coke, or metal may become wedged between the tyres of twin wheels and, as the tyres flex at each revolution, the casings may be abraded and damaged. Usually such material is picked up by the tyres directly from the road but it may also obtain ingress to the gap between the tyres by way of the hand holes in the wheels.

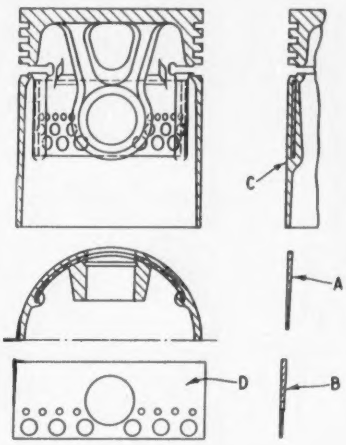
To obviate, or at least to reduce, this possibility it is proposed to clamp between the two wheels a flat disc A of a diameter to extend beyond the greatest section width of the tyres and to remain clear of the road surface at the maximum deflection of the tyres under ordinary operating conditions. The disc may be furnished around its periphery with a T-flange B of metal or a stiff resilient material and of a width to approach closely the tyres when undergoing maximum deflection, as when passing over a hump in the road. Lateral flanges C may be welded to each face of the disc at a suitable radial distance from the centre to prevent foreign bodies entering through the wheel hand holes and becoming trapped between the tyres.

In order to assist in the dissipation of heat from the brake drum and wheel centres, portions of the disc may be pressed out to form louvres D. The disc will then act as a fan, drawinp air from the wheel



No. 759048

centres and discharging it between the two tyres. If desired, the disc may be suitably proportioned to embody sufficient strength to prevent the complete collapse of the casings in the event of a double burst. The specification suggests that plain discs may be mounted outside the wheels to prevent damage by kerb-scraping, but means of attachment for such discs are not shown or described. *Patent No.* 759048. *Firestone Tyre and Rubber Co. Ltd.*



No. 759191

Bimetallic piston

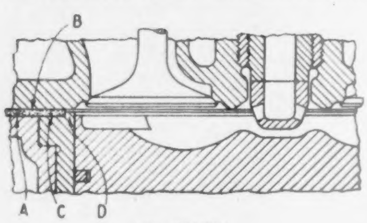
A well-established practice to control the thermal expansion of a light-metal piston is to incorporate inserts of a metal having a lower coefficient of expansion and exploit what is known as the "bimetallic" effect. However, since the expansion of a piston differs at different levels, it is desirable that the bimetallic effect is varied so that the piston at all times conforms to the cylinder bore and thus the clearance between piston and bore can be held to a minimum. In the slotted piston illustrated two inserts of steel strip material are embodied in way of the gudgeon pin bosses.

Effective control can be provided by any one of several methods, or by a combination of methods. The steel insert may be of a section tapering smoothly from the upper to the lower edge, as at A, or may be stepped, as at B. Conversely, the piston wall may be of tapered section, as at C and the insert of uniform thickness. In a preferred form, however, an insert of uniform section is provided with one or more rows of perforations, as at D. Such perforations may be of different sizes and different shapes and an inperforate zone may be retained at the upper portion of the strip. This construction has the advantage of adding the minimum weight to the piston. In all cases the inserts are apertured to register with the gudgeon pin bores and the upper edge, at least, is unbroken. *Patent No.* 759191. *Aluminium Laboratories Ltd.* (*Canada*).

Sealing diesel engine cylinder heads

The joint between the cylinder block and the cylinder head is usually sealed by a gasket and the edges of this gasket are exposed to the hot combustion gases in the cylinder spaces. Particularly in the case of engines of the precombustion chamber type, in which jets of burning mixture issuing from the chamber are directed towards the periphery of the cylinder bore, the gasket edge is liable to suffer premature deterioration. Constructions are known in which the gasket is wholly or partially enclosed in an annular recess in the cylinder head, but such constructions are expensive to produce and not suited to rapid production, since the joint surfaces are not continuously flat and cannot readily be finished by grinding.

In the construction proposed, both the cylinder block and the cylinder head have continuously flat joint faces, A and B respectively. The cylinder liner has an external flange at its upper end, which is seated in an annular recess in the cylinder block. Upper face C of the liner flange is machined to lie in the same plane as the joint face of the cylinder block and receives



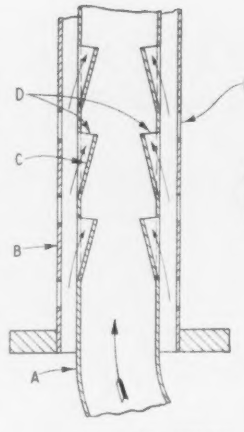
No. 761020

the gasket. Around the inner periphery of the flange, however, is an upstanding rim D, which extends towards face B on the head and is separated from it by only a production or thermal-expansion clearance, say 0·02 mm. This rim serves as a protective screen for the edge of the gasket. *Patent No.* 761020. *Daimler-Benz A.G.* (*Germany*).

Dilution of exhaust gases

Where internal combustion engines must be operated in confined or enclosed spaces the exhaust gases may be objectionable or injurious to personnel working in the same areas. Since the more noxious gases are less dense than air it has been proposed to discharge the exhaust gases at a height of not less than six feet above the floor so that they can rise and disperse above the level at which they would be inhaled. Gases can also be rendered less objectionable and less harmful if they are diluted before being discharged into the atmosphere. This invention provides a simple construction by which dilution can be effected.

The circular exhaust pipe A is disposed concentrically inside an open tubular member B. In the exhaust pipe is a series of pairs of diametrically opposed louvres C. Each louvre is formed by making a cut through an arc subtending 90 deg and depressing the upstream side of the cut so that it has an inward curvature equal but opposite to that of the pipe. Each pair of louvres thus forms a venturi-like constriction having two apertures D at its downstream end. The exhaust gas velocity is increased at each constriction, its pressure is reduced, and air is drawn into the gas stream through the apertures D.



No. 760493

Tubular member B is provided with a series of perforations E to permit the free access of air from the exterior. *Patent No.* 760493. *Coventry Climax Engines Ltd.*

Gear tooth form

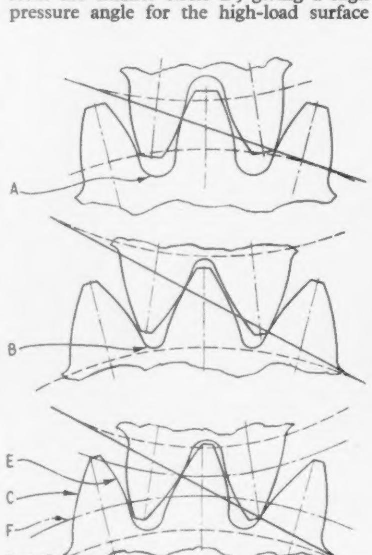
In normal involute gears, both tooth profiles are generated from the same base circle and pressure angles on opposite sides of the tooth are equal. Teeth of that type having a low pressure angle are characterized by a large fillet radius at the base, as at A, and insensitivity to variations in machining or mounting. Their load-carrying capacity, however, is restricted by reason of high bending moments and high contact stresses imposed on the individual tooth during operation.

By increasing the tooth pressure angle the high bending moments will be substantially decreased. The advantage of this practice, however, is accompanied by an undesirable decrease in radius of the tooth base fillet, as at B.

It is proposed, therefore, that opposite profiles of a tooth be involutes of different diameter base circles. Profile C is generated from the smaller circle D, giving a high pressure angle for the high-load surface in the normal forward direction of rotation. Profile E is from the larger circle F and has a lower pressure angle and a lower capacity for operation in the reverse direction. The teeth of the meshing gear are formed in a similar manner, and the high pressure angle profile of one engages the high pressure profile of the other, as shown.

The resistance to fracture on the high pressure angle profile, it is stated, is of the order of 30 to 40 per cent greater than that of a normal symmetrical tooth having equal pressure angles on both profiles. This additional beam strength is obtained without appreciable reduction from normal on the low pressure angle profile.

A further advantage of gears of this type is that the individual teeth are more flexible than conventional teeth of equal capacity and, therefore, can deflect more under load to adjust automatically for greater variations in manufacturing tolerances and misalignment. The invention is equally applicable to helical gears, since both are cylindrical in form. *Patent No.* 760091. *Caterpillar Tractor Co.* (*U.S.A.*).



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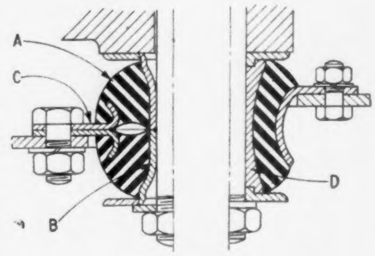

No. 760706

Resilient engine mounting

This double-acting mounting provides a high measure of resiliency under static load conditions and an increasing measure of stiffness under conditions of oscillation. Upper and lower portions of the rubber mass are in a state of precompression under no-load conditions and the stress-strain diagram of the support is symmetrical. The two portions of the resilient mass act gradually in taking up the load and absorbing the rebound as a result of progressive deformation of the mass under thrust from abutment surfaces on the central support member. Reference is made to an earlier patent, No. 697449.

Two examples are illustrated. In one a spherical resilient mass A is bonded to a tubular member B which, with its upper and lower abutment flanges forms the "central terminal member" for attachment to the engine casing. A double-flanged member C, embedded in the resilient mass forms the "outer terminal member" and is secured to the vehicle framing. The axial-section contours of the two terminal members are such that both upper and lower portions of the mass are subjected first mainly to shear-flexion stress and then mainly to compression stress. Passage from the one type of stress to the other is gradual.

Tubular member B, it will be noted, is made in complementary halves and the degree of precompression on the resilient mass can be modified by adjusting their axial proximity. If desired, however, the central tubular member may be of unitary construction, as shown at D in the alternative example. *Patent No.* 760706. *Societa Applicazioni Gomma Antivibranti S.A.* (*Italy*).


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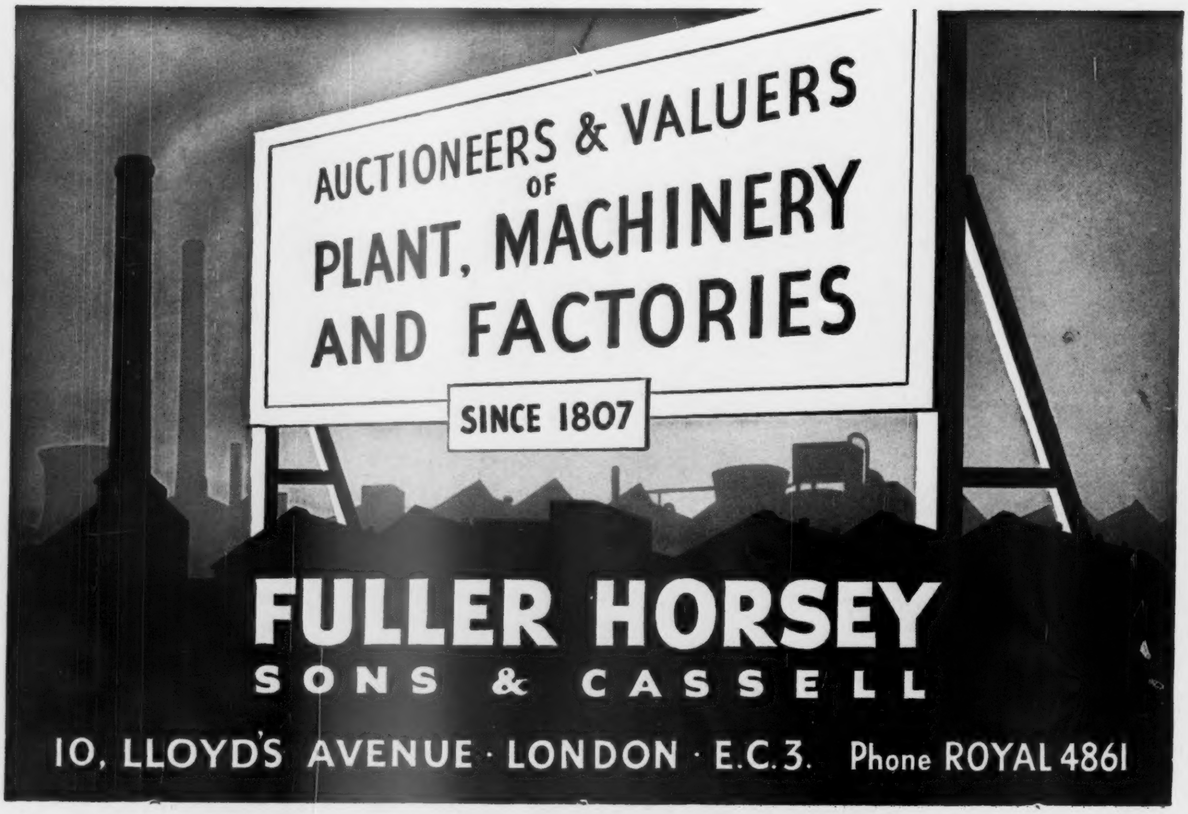

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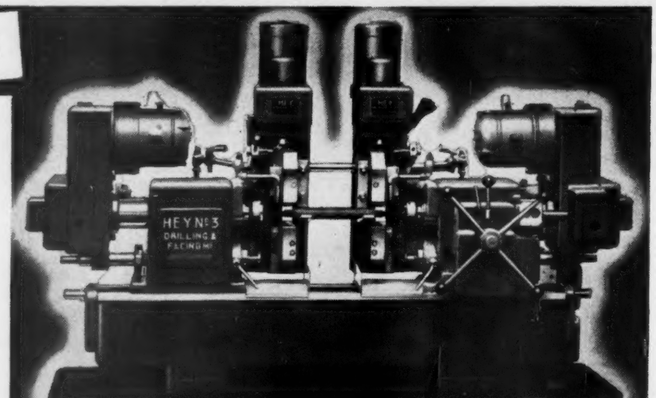

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SIMMS MOTOR UNITS LIMITED

SUMMARY OF MR. G. E. LIARDET'S STATEMENT FOR YEAR ENDED 31st DECEMBER, 1956

The Annual General Meeting of Simms Motor Units Limited was held on July 26th, 1957, at the Company's Works. Mr. G. E. Liardet (Chairman) presided and, in the course of his speech, said:—

"I do not think you will be unduly disappointed with the results, particularly if you remember my comments at this time last year that the costs of labour, material and all services were rising. Following these difficulties we were faced with the recession in the Vehicle Industry and then the Suez crisis. I feel it would be your wish for me to thank all Employees for the loyalty and sustained effort which has enabled us to achieve the results reported in these Accounts.

"Although the Group Turnover, excluding new acquisitions, shows an increase over 1955, the Trading Surplus has declined by £156,869 to a total of £1,040,547. This is due to rising costs of material and labour absorbed by the Company and the effect of credit and fuel oil restrictions upon the trading of certain Subsidiary Companies.

"After providing for Taxation and a transfer to Capital Reserve the balance of Profit available is £401,583 compared with £512,251 in 1955. General Reserve receives a further £150,000 and Plant Replacement Reserve an additional £100,000. Your Directors recommend an Ordinary Dividend of 1/9d. per share which, if approved, will leave a balance to be carried forward of £392,788.

"The re-equipment of the factory at Finchley has continued and some progress has been made with similar work at the Branches and Subsidiary Companies.

"Unless there is continuous replacement of plant with modern equipment our productive industries will be unable to meet foreign competition, and it is regrettable and very disappointing that in the last Budget concessions were not made to industry in general for this purpose. The present rate of Company taxation makes it increasingly difficult to effect such replacement from retained profits unless these bear a high ratio to turnover, in which case our products are again uncompetitive in overseas markets. It is vital that industry, and particularly export industry, shall retain sufficient profit for plant replacement.

A YEAR OF PROGRESS.

"During the year 1956 our Engineering Division was expanded in personnel, facilities and area. A number of new products are being developed and, while considerable expenditure is involved, we are confident that as a long-term measure it is fully justified. It is encouraging to find that we receive an increasing number of requests to solve problems relating to internal combustion engines and turbines.

"Our manufacturing Plant has been enlarged and improved and we estimate there are over 500,000 sets of fuel injection equipment in service on various engines in many parts of the world.

"Towards the end of the year we added two new projects to our range. The first is a Turbo-Charger for internal combustion engines, for which we concluded a licence with Geraetebau Eberspaecher O.H.G. of Esslingen, Western Germany; the second project is a Hand Starter for diesel engines, the designs and patents for which we acquired from Aircraft Steel Structures.

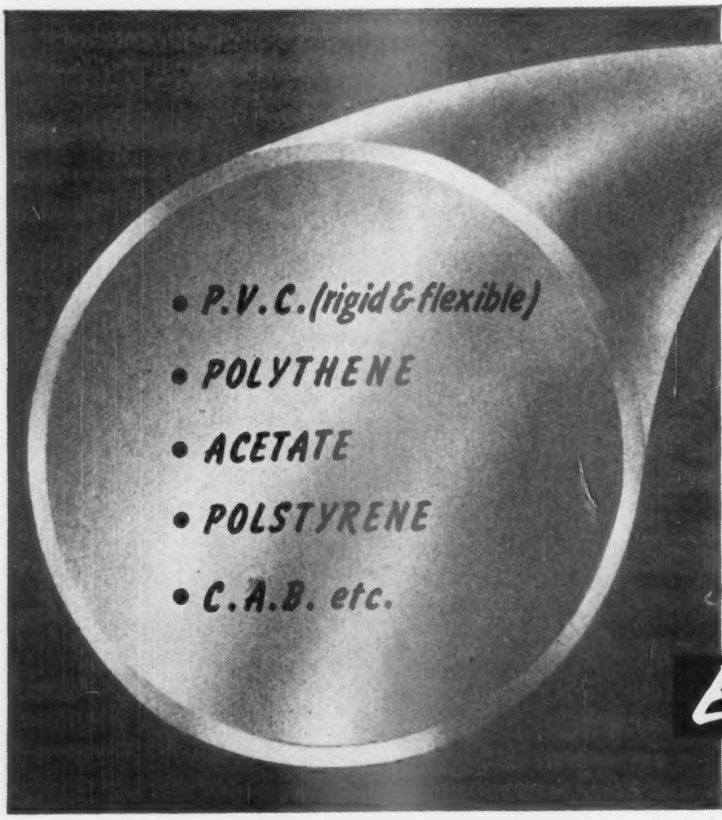
"I am pleased to report that our Export turnover reached a new high level, and our Home Branches performed well and made a very substantial contribution to Profit. You may be interested to know that during the year we set up new Departments at most of our Home Branches to provide a trade Repair service for all makes and types of motorcar electrical equipment, as distinct from our previous activities, which were almost entirely connected with commercial vehicles. The response to Simms' standards of quality, price and service was immediate. Our Repair organisation is already one of the largest in the country and we look forward to a big increase in this business.

"We are pursuing the policy of spreading the range of our products over a much wider field and we have accordingly added the Industrial Fan & Heater Company of Birmingham to the Simms Group of Companies, in addition to Mono-Cam Ltd. and R. F. Landon & Partners Ltd. to which I referred last year.

"In December we acquired by means of an exchange of Shares one-third of the issued Share capital of Aircraft Steel Structures Limited.

FUTURE.

"Unfortunately the future is somewhat obscure, for though your Company is in good heart, with an able and energetic team ready to seize every possible opportunity, yet if Trade Unions continue to insist on wages rising out of proportion to productivity and the Government continue to spend at a rate which requires penal taxation at a level which curbs initiative and depletes the resources of industry, then your Company is bound to be influenced by the national trend."



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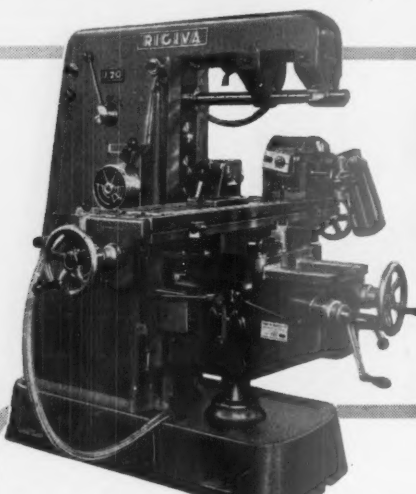

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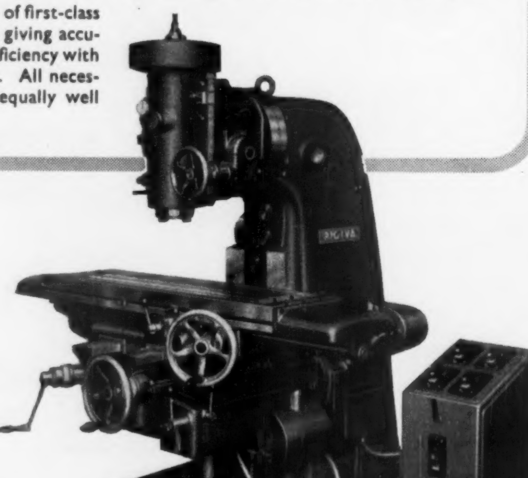

MODEL U 20

Universal; table working surface 55" × 11 13/16", power longitudinal traverse 43 3/8", cross traverse 13 3/4", table rise and fall 13 3/4", spindle nose 2 3/4", International Taper.

MODEL V20 B

Vertical; table working surface 55" × 13 13/16", power longitudinal traverse 43 3/8", cross traverse 16 1/16", table rise and fall 17 3/4", spindle nose 2 3/4" International Taper.

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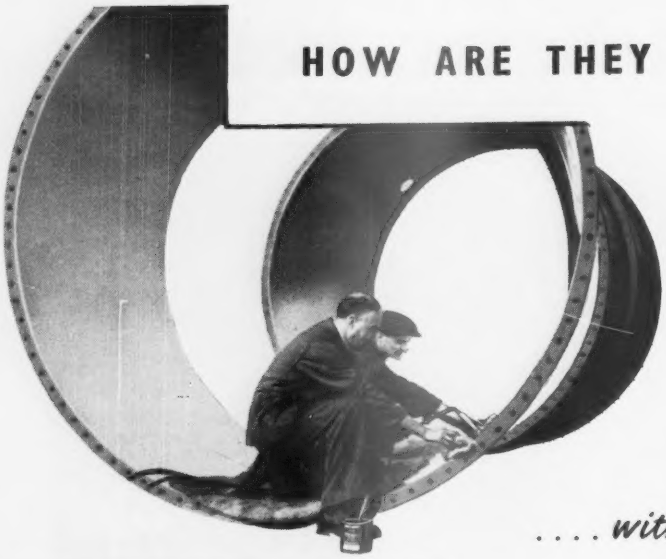

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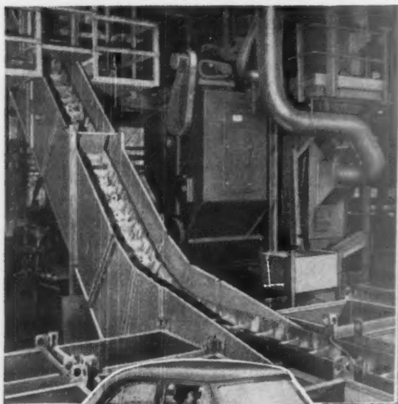
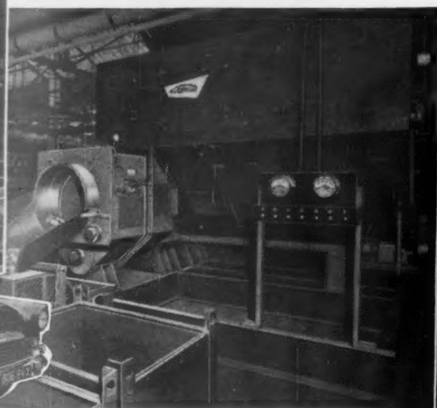
The illustrations :—
Top left : Loading conveyor.
Centre : General view, front of plant.
Lower right : Discharge drum and conveyor.





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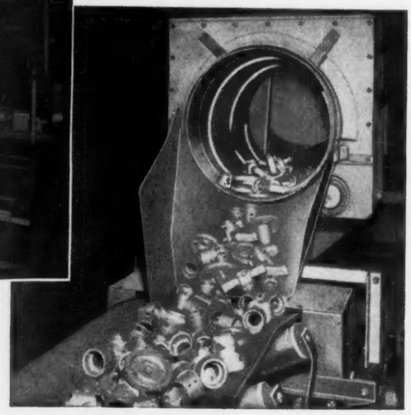

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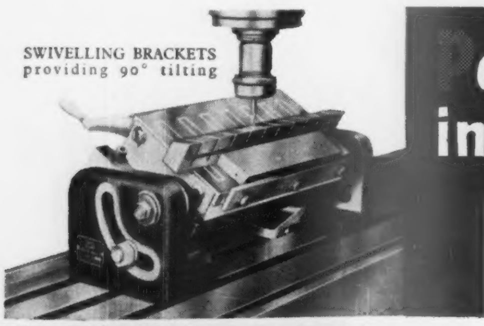

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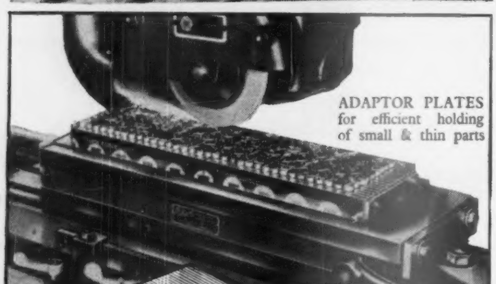

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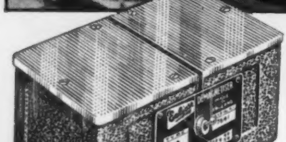

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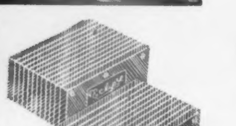

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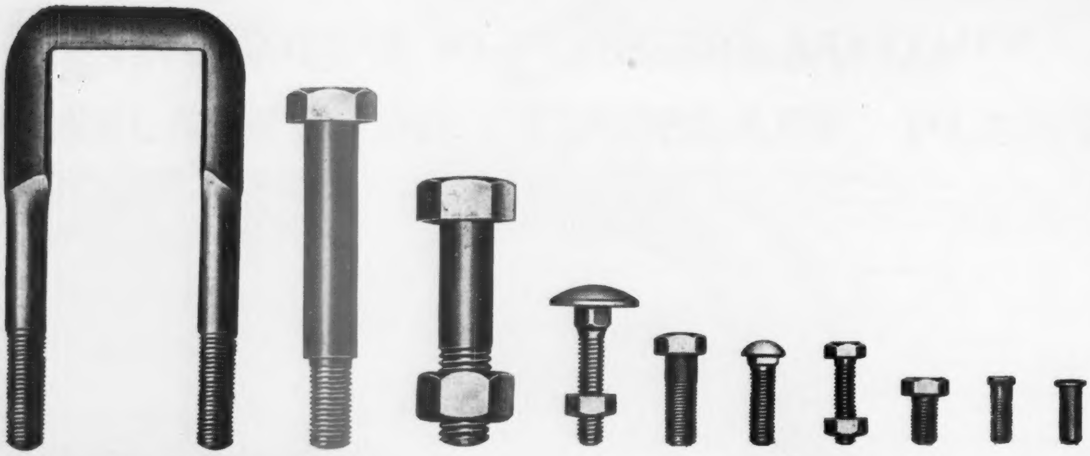

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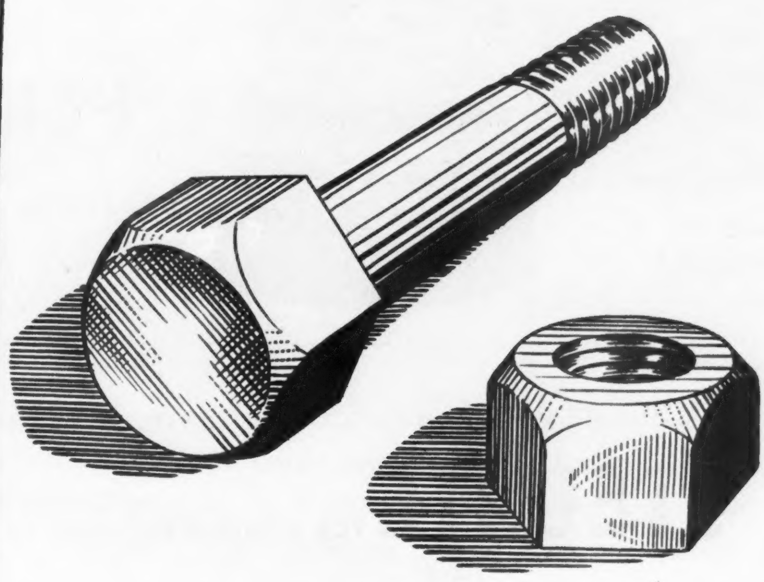

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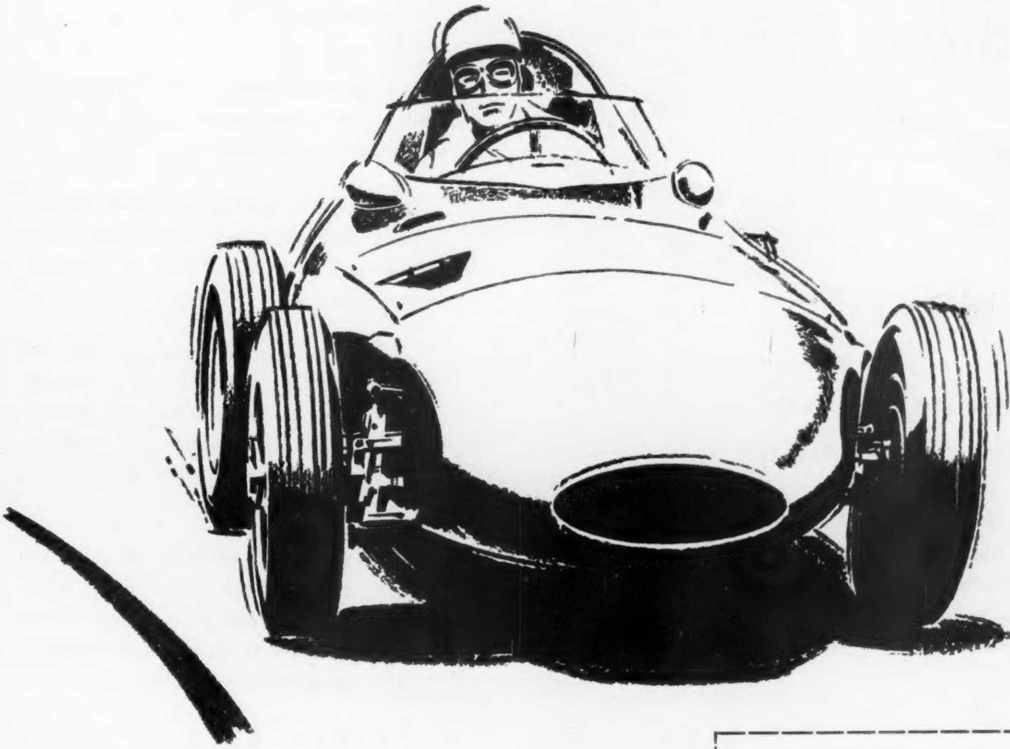

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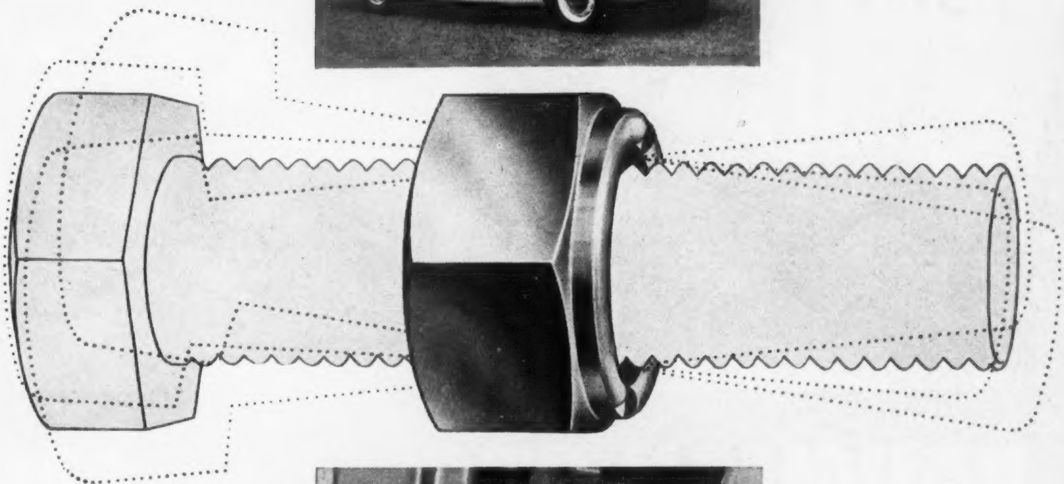

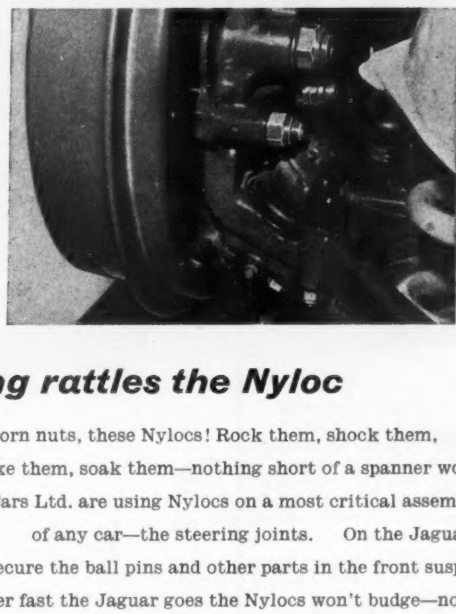

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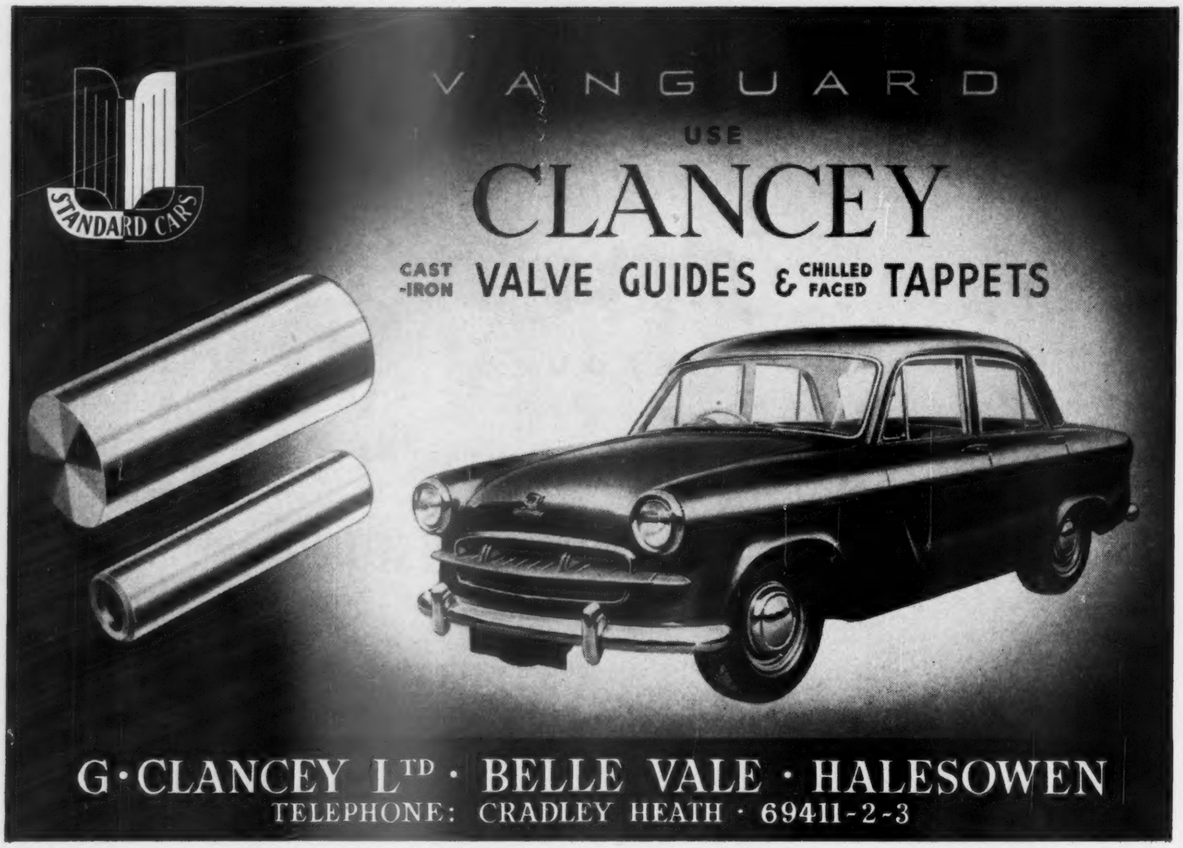
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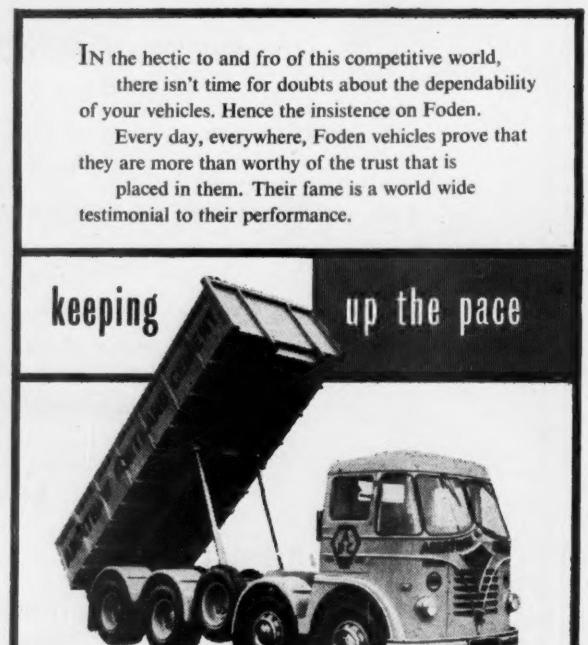
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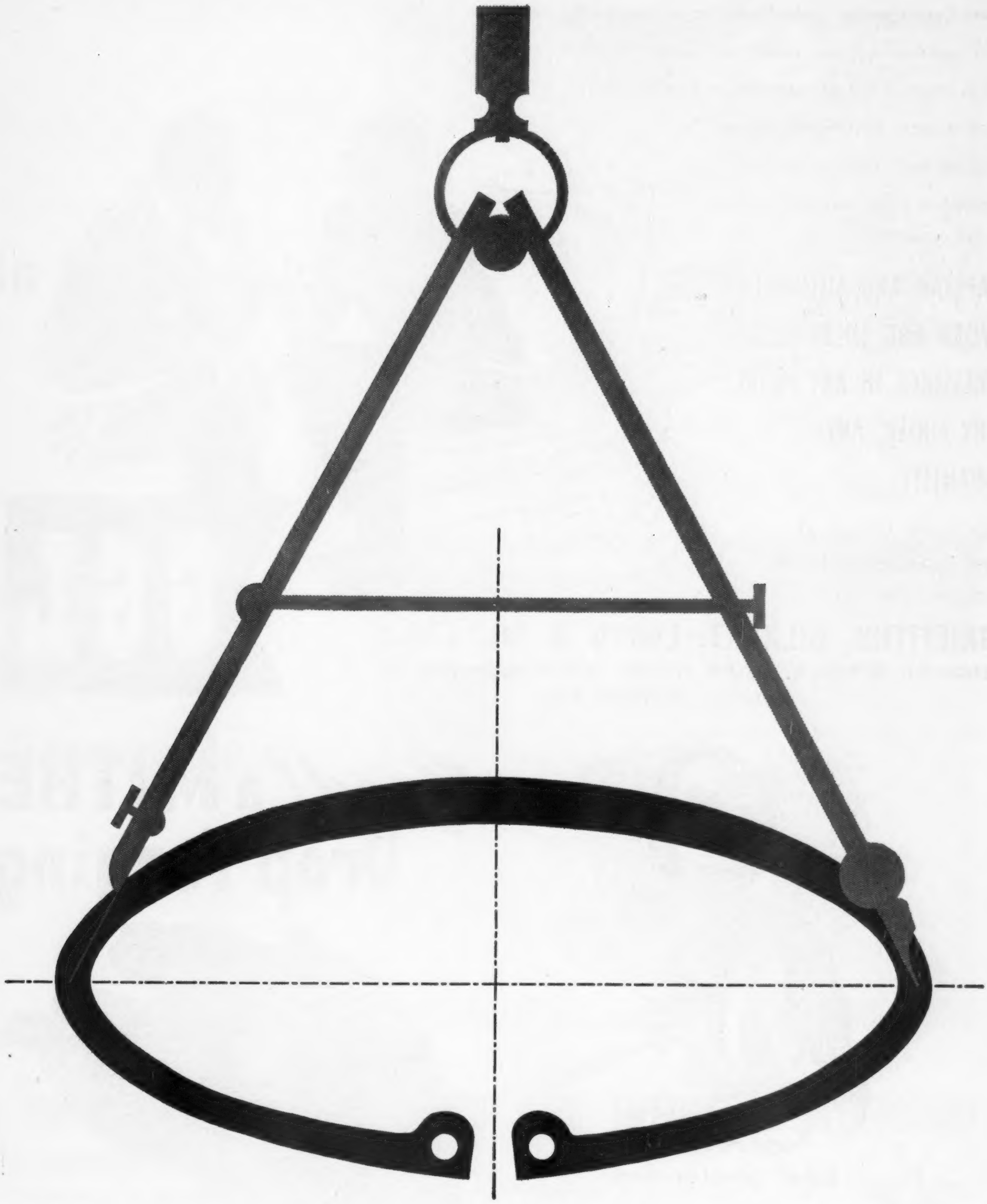

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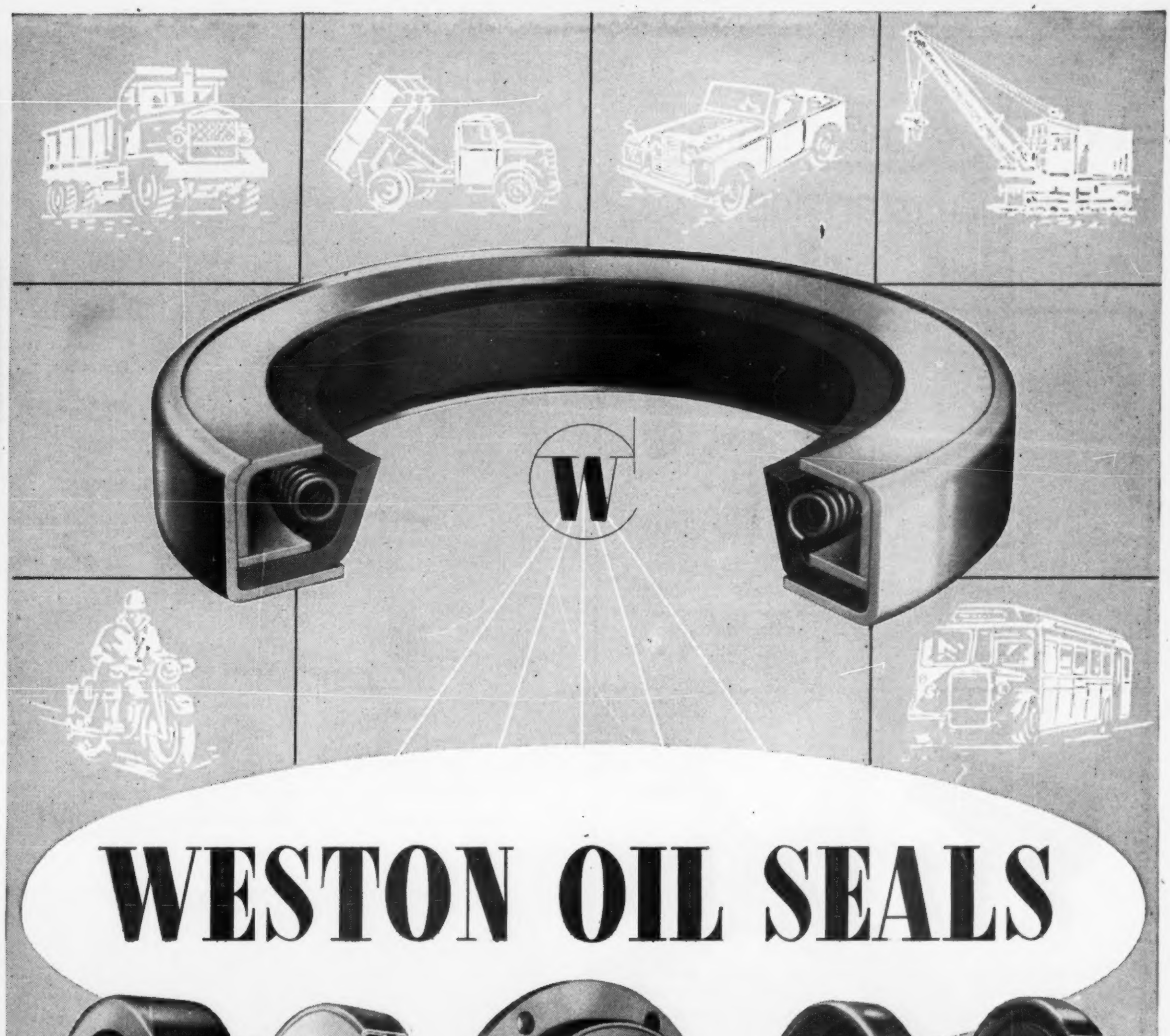
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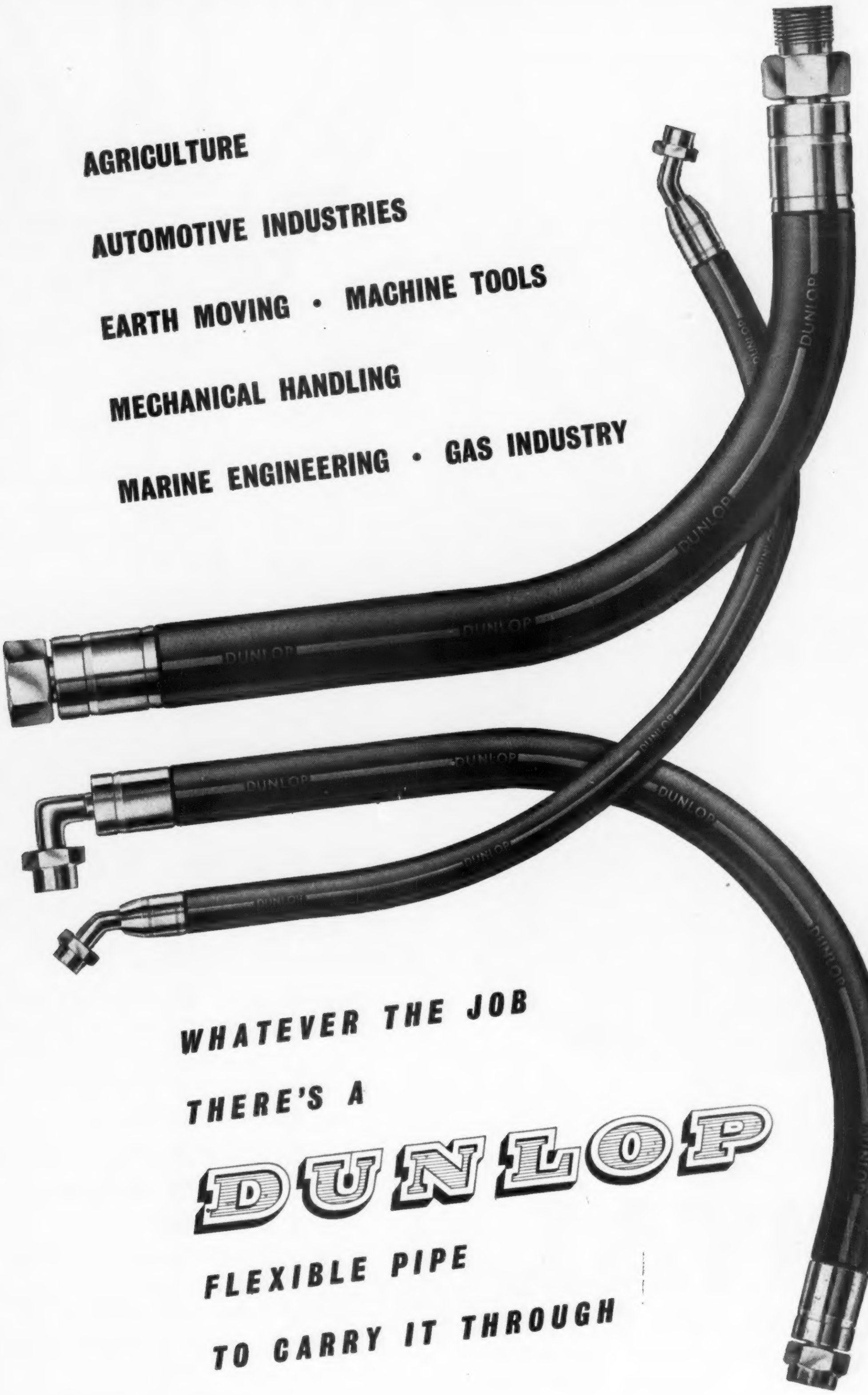

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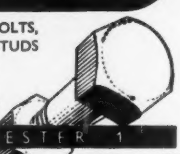

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